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CONTENTS

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Paper No. 1

ELECTROSLAG WELDING

by

R. MORRISON

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

ELECTROSLAG WELDING

By R. Morrison

INTRODUCTION

The successful introduction of the electroslag welding process has necessitated revision of some features regarded almost as standard practice in welding.

As there are, as yet, relatively few installations operating in this country it is thought that some notes on the basic principles and applications may be of interest.

Although applications may be varied to requirements of thickness, size and materials, the instances here are intended to convey a general picture of the procedure of welding by this process.

The electroslag welding process was developed in Russia and demonstrated as a production process in 1958 at Brussels.

A principal feature of the process is that welds are completed in a single run and the machine is readily adaptable to the welding of mild steel, low alloy steels or stainless steels. The technique is particularly applicable to thick sections where previously welding would be considered impracticable and components for severe duty services in nuclear installations and boiler steam-raising plant are now being manufactured on a regular production basis in this country by the process.

The Principle of Electroslag Welding

In welding by this process a pool of liquid metal across the entire thickness of the plates to be welded is contained by water-cooled copper shoes which slide as welding continues vertically upwards. The arrangement is shown diagramatically in Fig. 1.

The plate material to be welded is prepared with a square edge and set up vertically with a gap of $1\frac{1}{4}$ in.–$1\frac{3}{8}$ in., irrespective of plate or section thickness. The water-cooled copper shoes are located front and back of the gap and a "starting block" is fitted to seal the mould. One, two or three electrodes may be required depending upon the thickness of plate being welded. Initially the arc is submerged in powdered flux but after a period of about one minute the flux is converted to a highly fluid slag.

Electroslag welding commences when the slag bath reaches a certain depth and voltage and amperage are adjusted to extinguish the arc. The temperature of the slag is then maintained at 1750°–2000° C. by the passage of current from the tip of the electrode entirely across the liquid slag and weld metal fuses into the plate edges. The weld metal in contact with the copper shoes, which have a surface film of slag, solidifies and the shoes move slowly upwards with the carriage supporting the electrode. During welding the electrode is arranged to oscillate slowly across the weld between the shoes with a travel adjusted to arrest the electrode $\frac{1}{2}$ in. from the shoe, the rate of oscillation being about twelve strokes per minute. This ensures an even distribution of heat input and assists to maintain a uniform penetration on the cross-section of the weld.

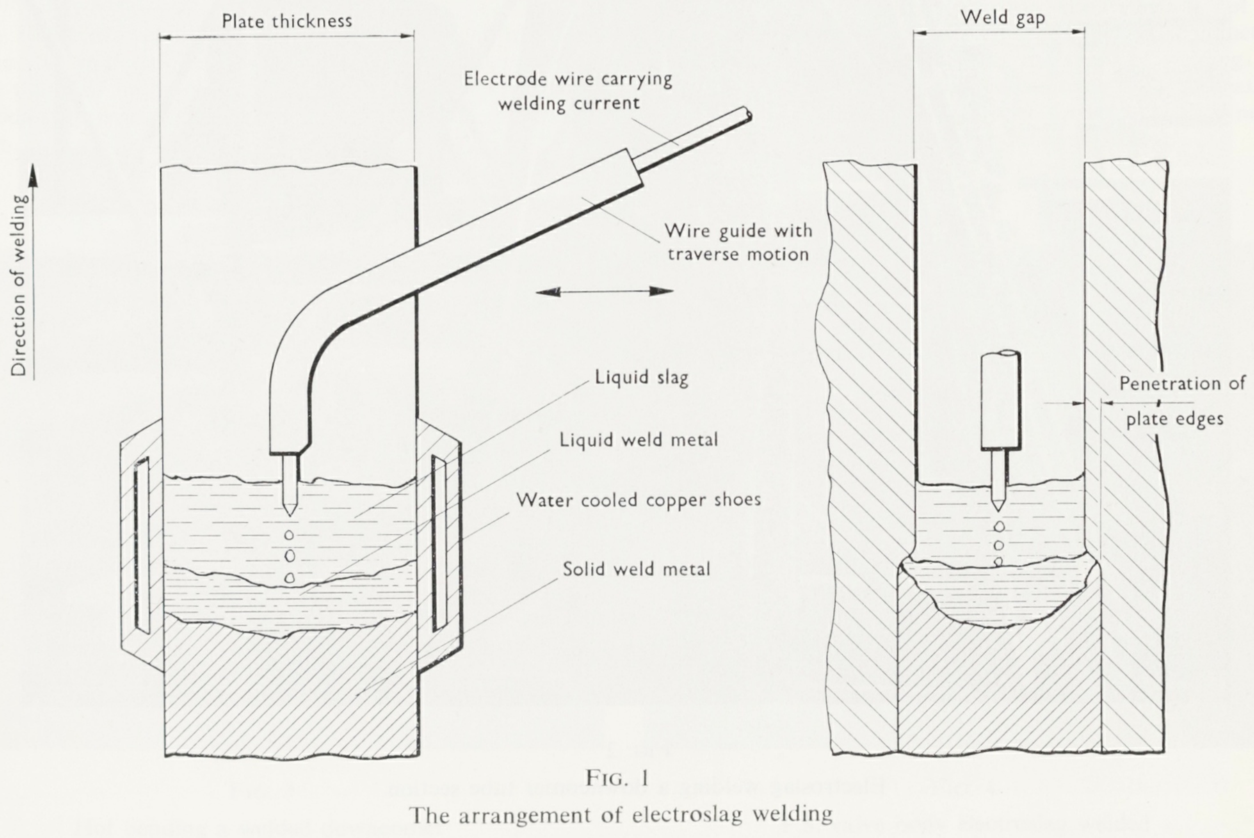


Fig. 1

The arrangement of electroslag welding

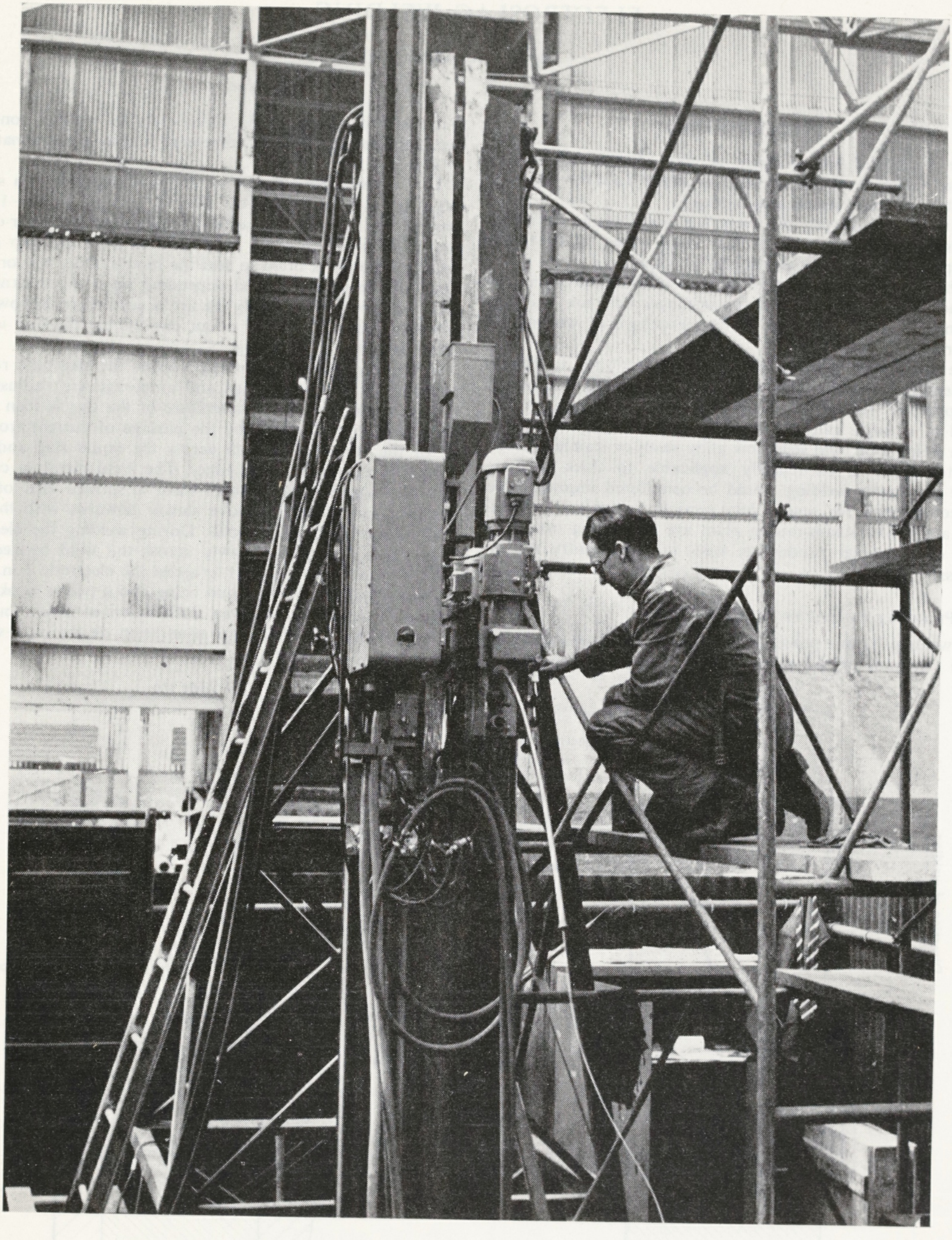


FIG. 2
Electroslag welding a downcomer tube section.

When equilibrium conditions are established the travelling carriage with the welding head and the sliding shoes move up the seam at an automatically controlled rate, keeping the level of the molten metal constant in relation to the electrode guides and the top edge of the copper shoes.

Electrode, Flux, Deposited Metal and Penetration Effect

The welding wires normally in use are copper-coated steel wire $\frac{1}{8}$ in. diameter, either solid or flux cored.

The wire used for the welding of mild steel by the electroslag process has a nominal 2 per cent manganese content, a typical analysis being:—

Carbon		0·12 per cent
Silicon		0·32 „
Manganese		1·96 „
Sulphur		0·023 „
Phosphorus		0·018 „
Nickel		0·15 „
Chromium	Less than	0·15 „
Molybdenum	Less than	0·15 „
Copper		0·30 „
Tin	Less than	0·02 „

The flux, of which there are a number of types, is of the usual powder form. It is chemically inactive during welding and has little influence on the metallurgical properties of the deposited weld. The flux is also required to electrically stabilise the process and this is achieved by the composition including chemicals to prevent arc formation. A flux with a high fluoride content is recommended.

Welding Speed

The speed of welding is, of course, dependent upon the thickness of material being welded and the number of electrodes that can be employed for a particular thickness. One electrode wire is used for thicknesses up to $4\frac{1}{2}$ in., from 5 in. to 9 in. two electrodes and thereafter three electrodes.

The following table is a general guide to the speed of electroslag welding.

Plate thickness	*Number of Electrode wires*	*Speed of Welding ft. per hour*
1 in.	1	8
2 in.	1	6
4 in.	1	3·5
8 in.	2	2
14 in.	3	2·5

The composition of the deposited metal, however, is not entirely that of filler metal and is, in fact, substantially influenced by the parent plate analysis, this being due to the effect of dilution.

Effects of Current Settings

The current settings require variations depending upon the materials but for the welding of mild steel the power, which may be either AC or DC, would be in the order of 550–600 amps., 40–50 volts.

Amperage is directly related to welding speed, an increase in amperage producing an increase in speed of welding.

Voltage regulates the degree of penetration and requires fine adjustment, an increase in voltage producing an increase in penetration. The contour of the outer surface of the weld is, of course, controlled by the inside surface of the sliding shoe.

Preheating

Due to the high energy input per unit length preheating of thick sections of mild steels and low alloy steels is not considered to be necessary as the effect of thermal conductivity of the plate being welded precedes the speed of welding. This feature of the process is often referred to as "self preheating". However, where the sections are of appreciable thicknesses, preheating of the starting area is recommended.

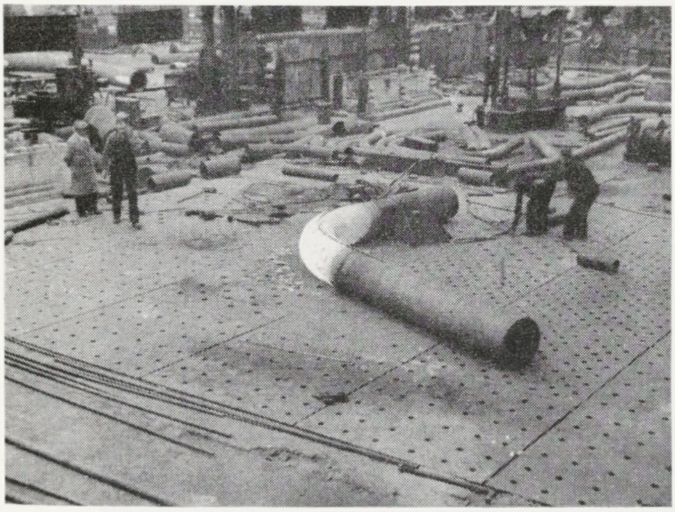


Fig. 3

Hot bending a welded downcomer

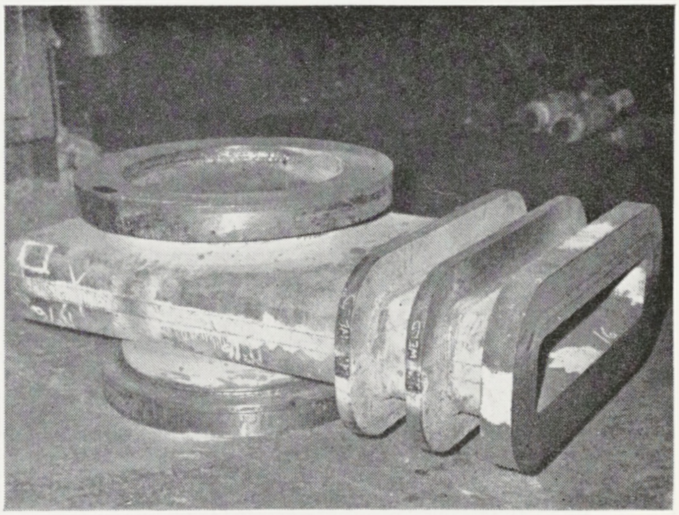


Fig. 4

P.S. valve body electroslag welded

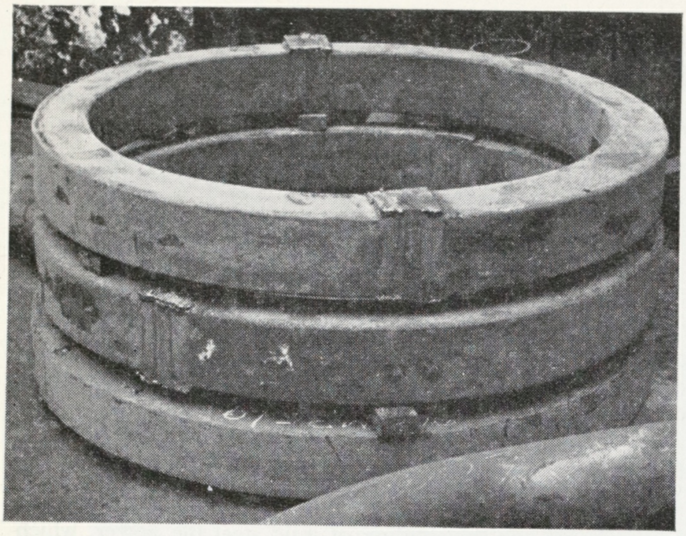


FIG. 5
Electroslag welded flange rings $8\frac{1}{2}$ in. square section

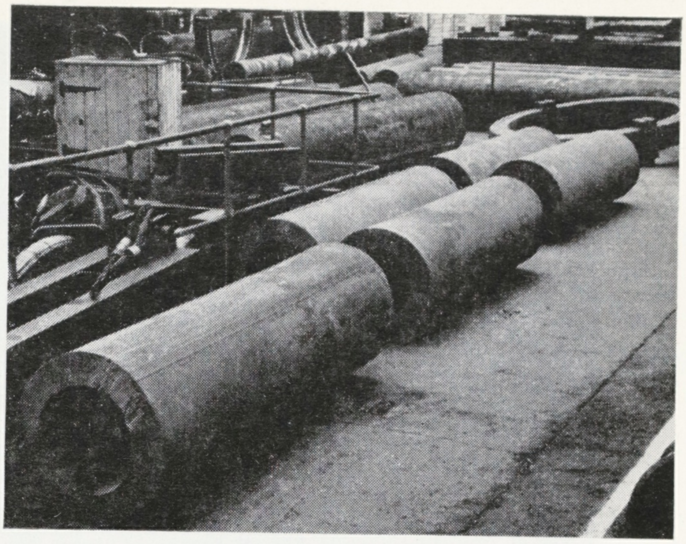


FIG. 6
12 in. bore charge nozzles

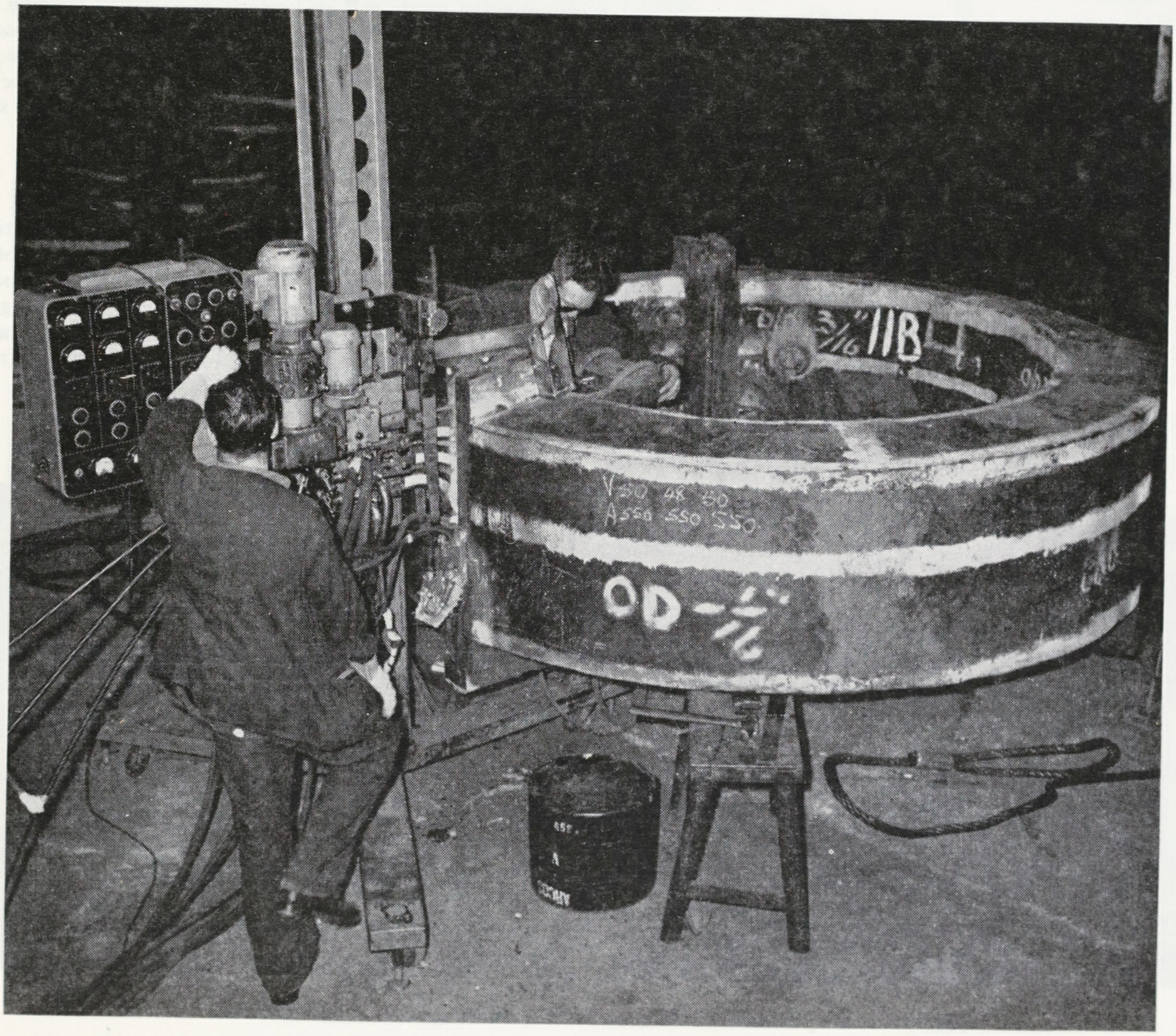


FIG. 7
Electroslag welding a 14 in. thick seam of a CO_2 nozzle

PRODUCTION APPLICATIONS

An early production application of the process was the fabrication of downcomer sections for power station boilers where operating pressures of 2,750 lb. per sq. in. are usual. The downcomer sections are about 18 in. inside diameter and vary in thickness from $1\frac{1}{4}$ in. to $3\frac{1}{2}$ in. Previously these would be ordered as seamless tube. Fig. 2 shows plate sections pressed to form a cylinder about 12 ft. long and the assembly during welding of the longitudinal seam. The welding of the longitudinal seam or seams by the electroslag process does not present any difficulties.

Fig. 3 shows the assembly of fabricated downcomer sections, 18 in. diameter, $2\frac{1}{2}$ in. thick, on completion of hot bending after conventional welding of the circumferential seams.

Fig. 4 shows the fabrication of the body for a 22-in. bore parallel slide valve, the plate material being $1\frac{1}{4}$ in. thick. Each body has two longitudinal seams and special shoes were manufactured as the sides with the weld seams were only 6 in. wide. Thirty-two valve bodies were manufactured by this method.

The fabrication of large rings for use as flanges is shown in Fig. 5. Two forged quality sections $8\frac{1}{2}$ in. square are pressed to shape and the welding of two seams in each ring completed.

Some five hundred charge nozzles for the reactor of a nuclear power station are at present in production. The sections are 8 ft. in length and vary in thickness from 4 in. to 5 in., the assembly consisting of two plates pressed to form an internal diameter of only 12 in. Sections on completion of welding are shown in Fig. 6.

Another application of the process is the fabrication of the CO_2 nozzles for the reactors of a nuclear power station. Fig. 7 shows the assembly of three sections of forging quality material pressed to form a ring 8 ft. $8\frac{1}{2}$ in. outside diameter, each weld being 14 in. thick and 2 ft. 6 in. long. Figures 8 and 9 indicate the arrangement at stages during welding, where it will be noted that three electrode wires are used in this instance.

The material used for all the components described is silicon killed mild steel having an ultimate tensile strength in the range of 28–32 tons per sq. in.

SOLIDIFICATION

Due to the effect of the large reservoir of heat the rates of solidification and cooling are slow and as a result the grain structure is coarse.

In the "as welded" condition, it can be seen from Figs. 10 and 12 that the weld has a structure of extremely coarse crystals in the periphery surrounding an area of relatively coarse equiaxed grains.

Fig. 14 is a macro of a longitudinal section of a weld indicating the "as welded" columnar grain structure with the dendrites curling upwards.

A macrosection indicating the various microstructures found is shown diagramatically in Fig. 15. Although the overheated zone is quite broad due to the high heat input during welding the coarse grain structures of the deposited metal and the adjacent overheated zone do not have any marked influence on tensility or ductility with material having the analysis as shown on page 10.

The results of impact tests can satisfy the requirements of most specifications but are, of course, improved by normalising to produce fine grain structures.

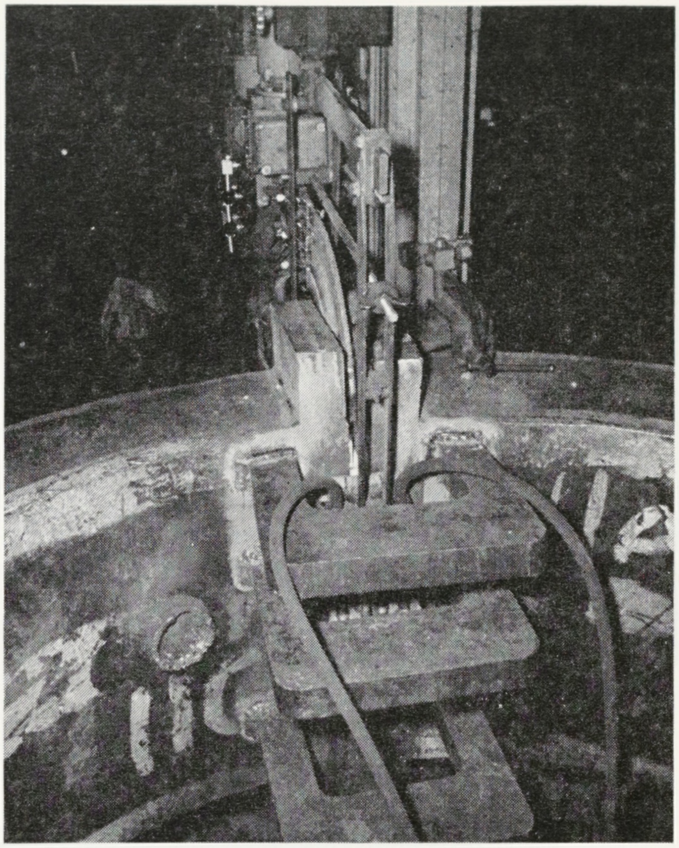


FIG. 8

View on the inside surface of a 14 in. thick seam during welding

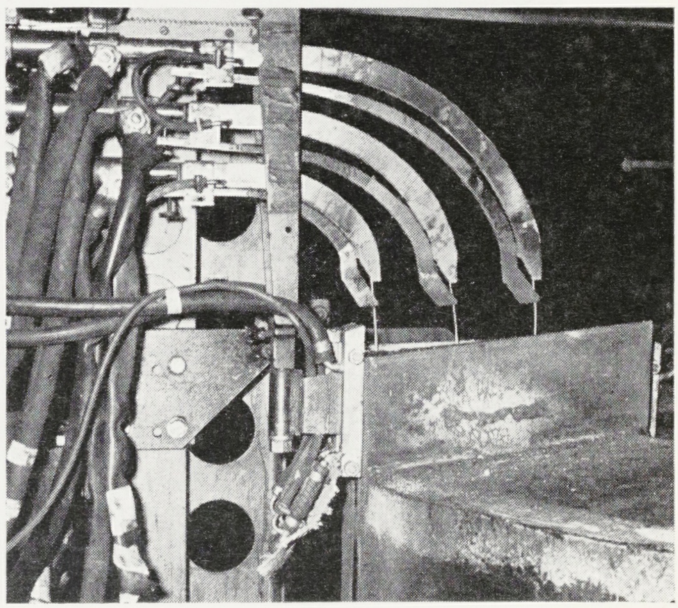


FIG. 9

Electroslag welding of a 14 in. thick seam completed

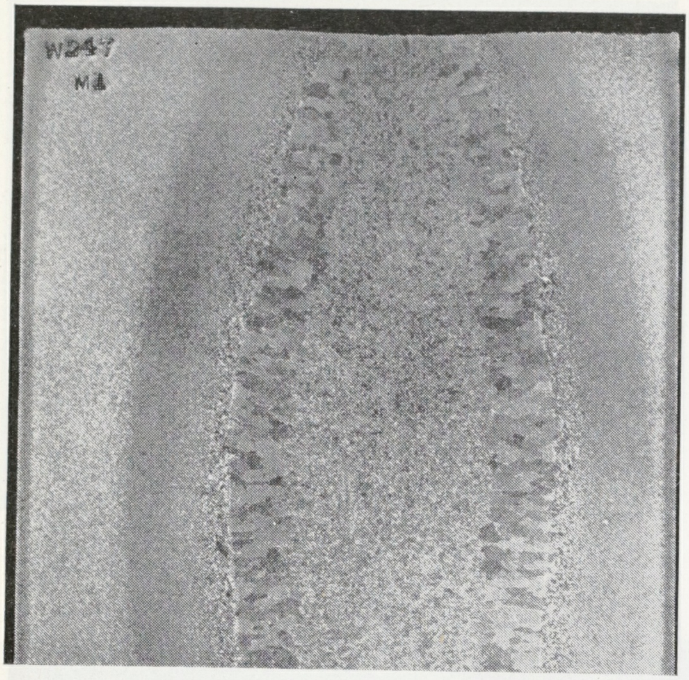


Fig. 10
Macrostructure of a $7\frac{1}{2}$ in. electroslag weld before normalising

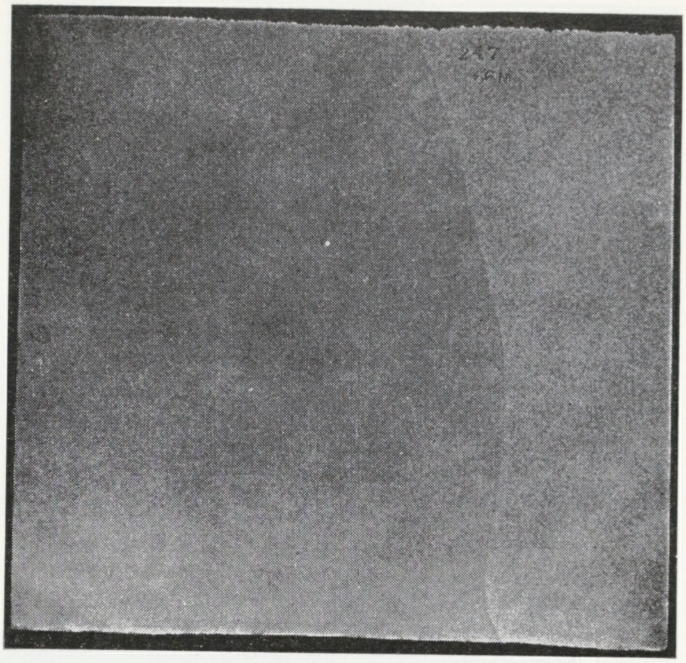


Fig. 11
Macrostructure of a $7\frac{1}{2}$ in. electroslag weld after normalising

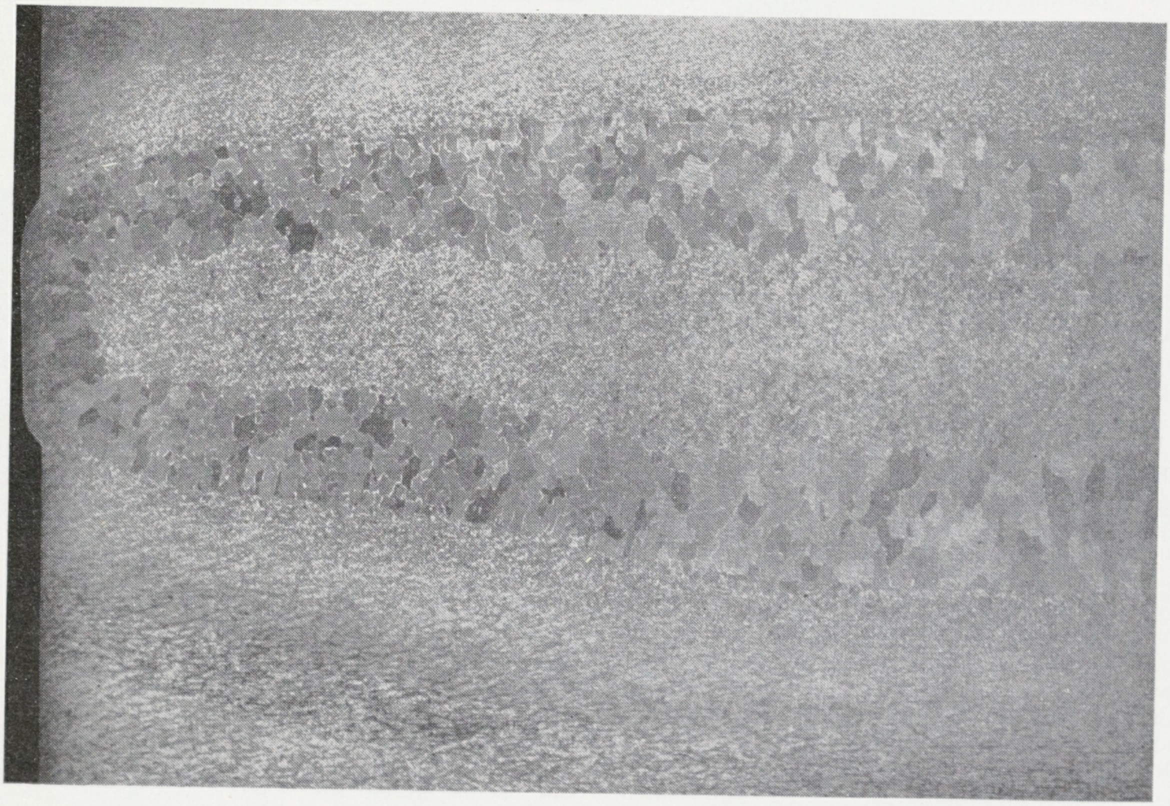


Fig. 12
Macrostructure of the half section of a 15 in. thick electroslag weld before normalising.

Following welding, the slow cooling through the transition zone although not an advantage with respect to grain microstructure of the weld, significantly prevents the formation of hard zones prone to cracking which can be a problem particularly with some ferritic alloys.

The process being such that the rate of weld progress is relatively slow and the weld contraction uniform, substantially reduces the residual stresses in the weld, which in unstress relieved conventional welds are at least as high as the yield point of the material.

Heat Treatment

For the severe duty applications of pressure parts for nuclear installations and boiler steam-raising plant it is at present considered desirable to refine the coarse "as welded" grain structure.

Refinement is obtained by a normalising heat treatment, which, in the case of mild steel, is within a temperature range of 900° C.–950° C. Figs. 11 and 13 show the macrostructures of welds on completion of heat treatment in the normalising range.

The microstructure of the electroslag weld after refinement by normalising is shown in Fig. 16 where the weld grain size is now reduced to that comparable with the plate material, Fig. 17.

As a further comparison Figs. 18 and 19 show the microstructures of submerged arc welds, single pass and multipass techniques, respectively.

All the microstructure photographs are taken at 100 magnification.

Weld Quality

The slow cooling and solidification also allows sufficient time for any slag to separate and for gases generated to escape from the molten pool. The welds therefore are, in general, free from slag inclusions and porosity.

However, excessive gassing due to laminations at a plate edge can cause "wormholes" in the weld, due to globules of gas being trapped in the solidifying metal.

Where, however, weld faults do occur due to mechanical or electrical failures in the machine during welding they are normally large defects with slag extending through the entire weld thickness. This is the most usual defect encountered and is readily visible on the outer surfaces.

Fig. 20 indicates the condition of the weld where the machine has been stopped and also shows the penetration of the plate edges. Continued welding of the seam would, of course, result in a defect on re-starting. In the event of a machine failure stopping welding it is necessary to restart within seconds, before solidification of the slag occurs, if a major defect is to be avoided.

Fig. 13

Macrostructure of the half section of a 15 in. thick electroslag weld after normalising.

Weld Repairs

Weld repairs fall into two categories depending upon whether the electroslag weld is in the "as welded" or the normalised condition.

Firstly, if the seam is in the "as welded" condition a high manganese mild steel electrode is required for the repair where the parent material is mild steel. The electrode deposit should match the analysis of the electroslag weld and maintain its physical properties on completion of the normalising treatment which the weld is subsequently to receive.

Secondly, if the defect is to be repaired after the seam has been normalised then a conventional repair using a mild steel basic coated electrode is satisfactory since the physical properties of hand electrodes are acceptable for a stress relieving treatment.

Non-Destructive Testing

Radiography applied to the electroslag welded longitudinal seams of downcomer sections has detected very few faults. In fact, the manufacturer states that 10,000 ft. of downcomer seam welded by the process in 1961 had only eight faults. The radiograph in Fig. 21 indicates an area of incomplete fusion in a $4\frac{1}{2}$-in. thick seam.

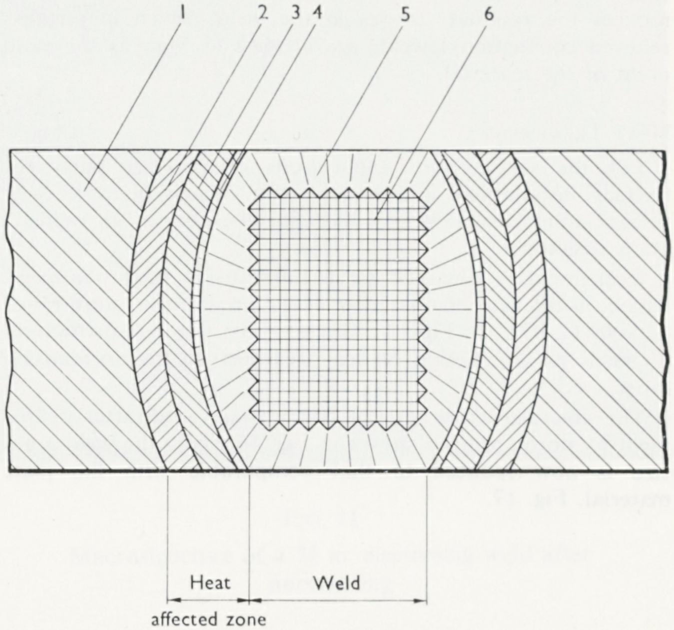


Fig. 15

Microstructures "as welded"

1. Plate material.
2. Zone of incomplete recrystallisation.
3. Zone of complete recrystallisation.
4. Overheated zone.
5. Weld metal of coarse structure.
6. Weld metal of equiaxed structure.

Fig. 14

Macrostructure of a longitudinal section of an electroslag weld before normalising

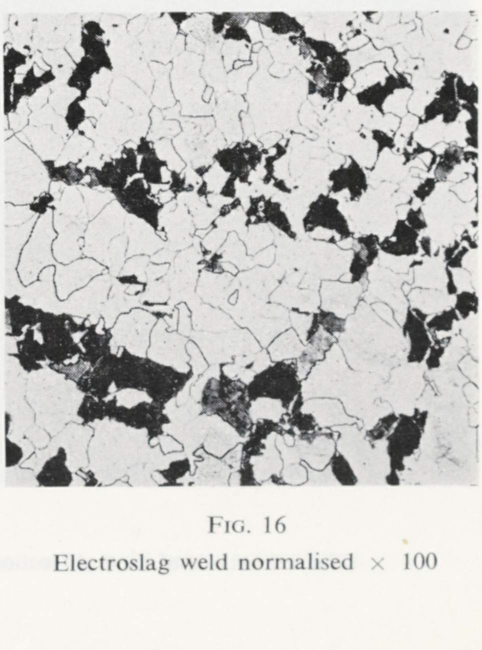


Fig. 16

Electroslag weld normalised × 100

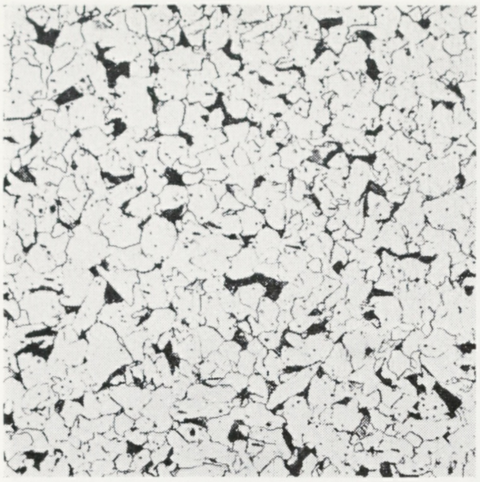


FIG. 17
Plate material ×100

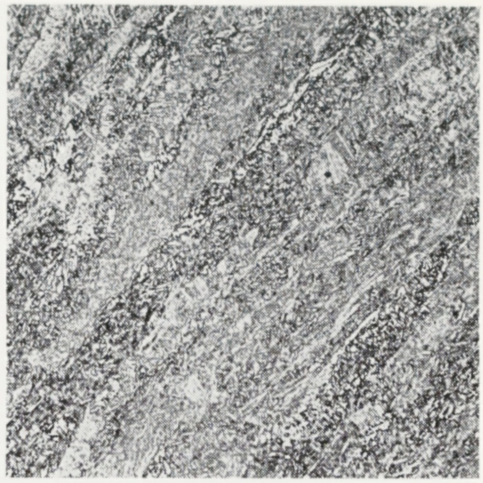


FIG. 18
Submerged arc single run technique ×100

Ultrasonic examination has been applied extensively to production welds made by the process in materials ranging in thickness from $2\frac{3}{4}$ in. to 14 in. Alternatively, radiography of thicknesses of this upper limit would require the use of either a linear accelerator or betatron equipment.

The welds are ultrasonically examined in two directions, through the length of the weld and through the thickness of the weld by manual operators. This method ensures the detection of any significant defect in the weld.

Welds made by the process should be normalised before being subjected to ultrasonic examination as the large grains existing in the "as welded" condition cause attenuation of the ultrasonic waves making interpretation of the oscilloscope trace difficult.

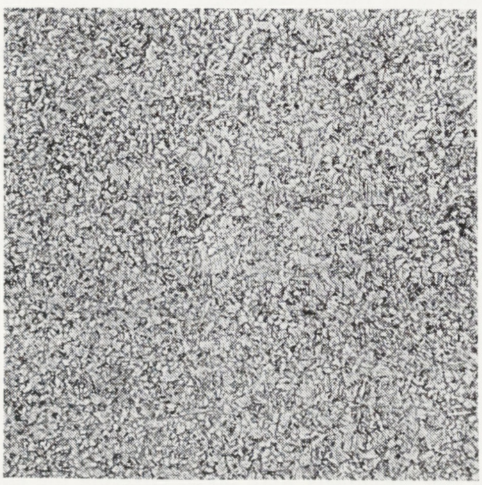


FIG. 19
Submerged arc multi-run technique ×100

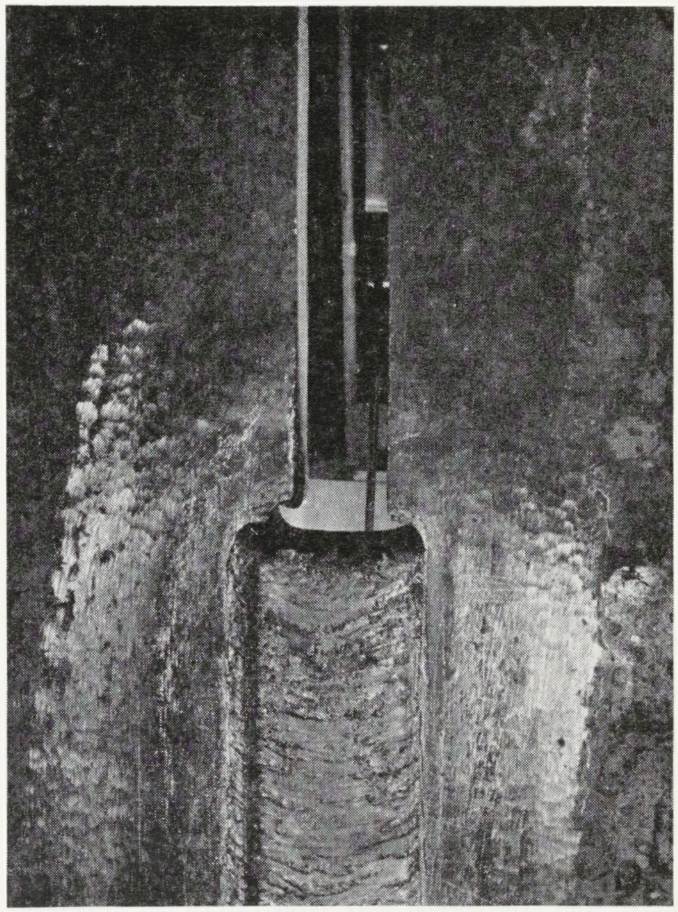


FIG. 20
The result of a stoppage in electroslag welding

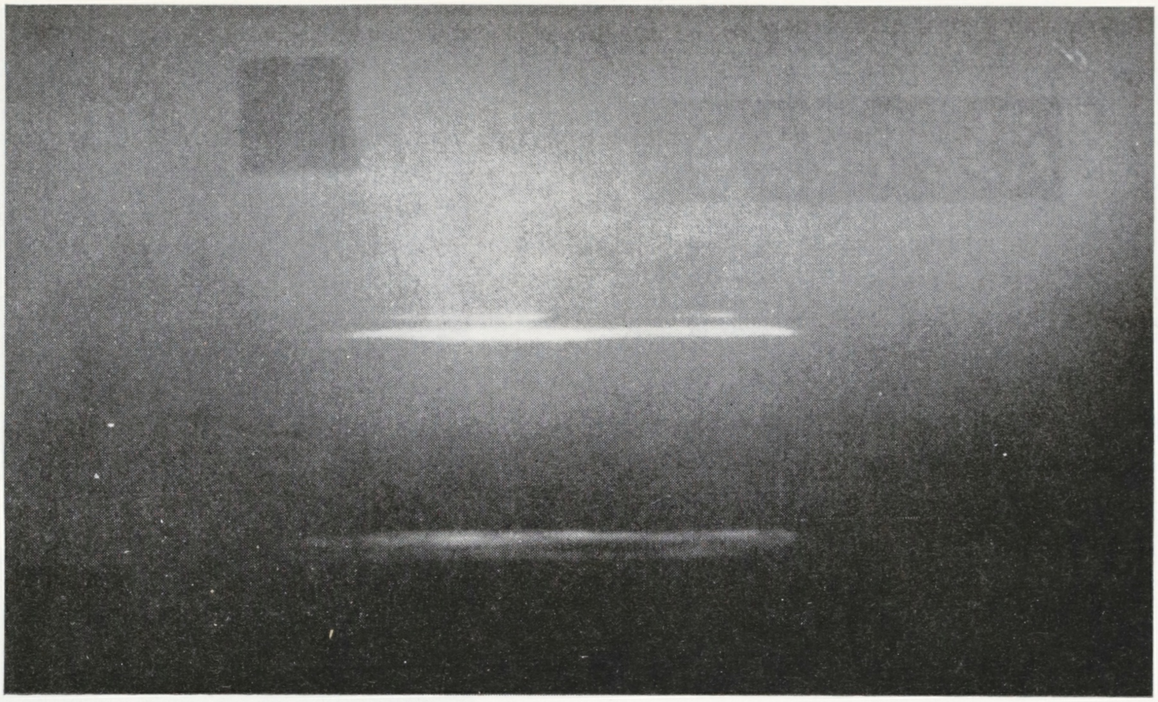


FIG. 21
Radiograph of a $4\frac{1}{2}$ in. electroslag weld with incomplete fusion.

Boiler Drums and Pressure Vessels

The welding of the longitudinal seams of boiler drums and other pressure vessels does not present difficulties and problems in the welding of circumferential seams can be overcome by maintaining a stationary welding position with the shoes vertical, turning the drum and adopting a suitable arrangement for leading the weld pool clear of the shell at the closure of the seam.

The heat treatment of large boiler drums, however, can present difficulties due to distortion occurring at normalising temperatures, a consideration of much less importance during the stress relieving treatment required for conventional submerged arc welds.

The approach to this problem here, however, is more likely to utilise a combination of the electroslag process and the submerged arc process. By electroslag welding the longitudinal seams, the shell sections can be normalised and re-adjusted for circularity and by welding the circumferential seams by a conventional process, the completed drum can then be finally heat treated by stress relieving.

The future situation concerning the requirements of heat treatment for the welds of the electroslag process is closely linked with experimental work being carried out to develop a technique to produce welds having a grain structure sufficiently fine to be acceptable for severe duty services without a normalising heat treatment.

Mechanical Properties

The quality of welded joints in thick sections can be represented by the results of the mechanical tests on a weld joint of sections $13\frac{1}{2}$ in. in thickness.

The material is mild steel forged quality slab and the tests are carried out after a normalising heat treatment.

Chemical analyses of parent material and the weld metal are:—

	Parent material	*Weld metal*
C%	0·13	0·13
Si%	0·27	0·21
Mn%	1·29	1·48
S%	0·007	0·012
P%	0·009	0·017
Ni%	0·16	Less than 0·15
Cr%	Less than 0·15	Less than 0·15

Microsections

Microsections taken at the locations indicated in Fig. 23 are shown in Fig. 22.

Bend Tests

1. Bend test specimens were taken covering the entire weld thickness as shown in Fig. 23. Fig. 24 indicates the bends completed with former having a radius equal to one and a half times the thickness of the specimen.

2. Fig. 25 indicates a rather exceptional bend test across the section of full weld thickness, the bending being carried out at a metal temperature of –10° C. in this instance.

Tensile Tests

1. The results of the transverse tensile specimens shown in Fig. 26 are as follows:—

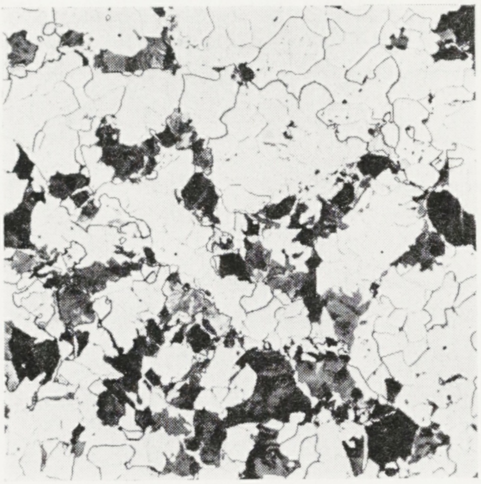
(*a*) 31·9 tons/sq. in.
(*b*) 32·5 tons/sq. in.
(*c*) 32·9 tons/sq. in.
(*d*) 32·3 tons/sq. in.

2. The results of the all-weld metal tensile specimen 0·564 in. dia. are:—

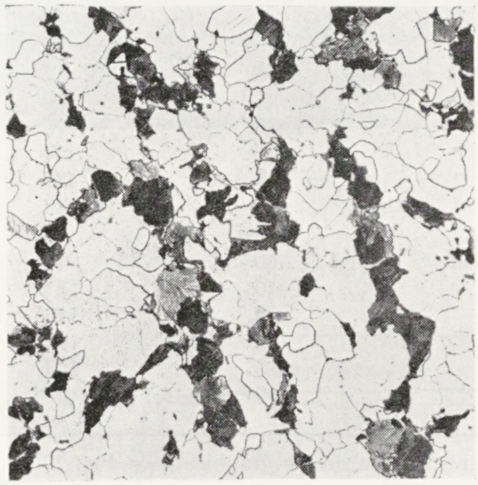
Yield point	20·4 tons/sq. in.
Tensile strength	32·6 tons/sq. in.
Elongation on 2 in. ...	30 per cent
Reduction of area ...	42 ,,

Notch Impact Tests

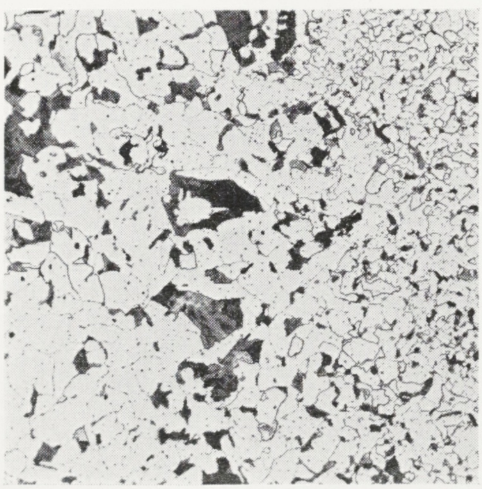
The results of the Charpy V–Notch Impact specimens as positioned in Fig. 23 are as follows:—



Weld centre × 100



$\frac{1}{4}$ in. from weld centre × 100



Junction × 100

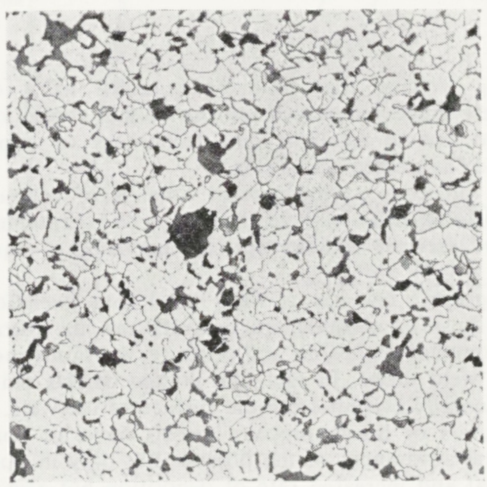


Plate material × 100

Fig. 22

Location	Testing Temperature °C	Charpy Values ft. lbs.
Weld Metal Centre	-60	7
	-40	12
	-20	16
	-10	22, 29, 29
	0	30
	20	50
	40	69
Weld Metal $\frac{1}{8}$ in. from Weld/Plate Interface	-60	9
	-40	27, 9, 16
	-20	14, 30, 32
	-10	39, 45
	0	38
	20	69
	40	88
Heat Affected Zone	-60	17
	-40	100
	-20	176
	0	176
	20	183
	40	187
Plate Material	-60	38
	-40	87
	-20	116
	0	164
	28	167
	40	171

The results are shown graphically in Fig. 27

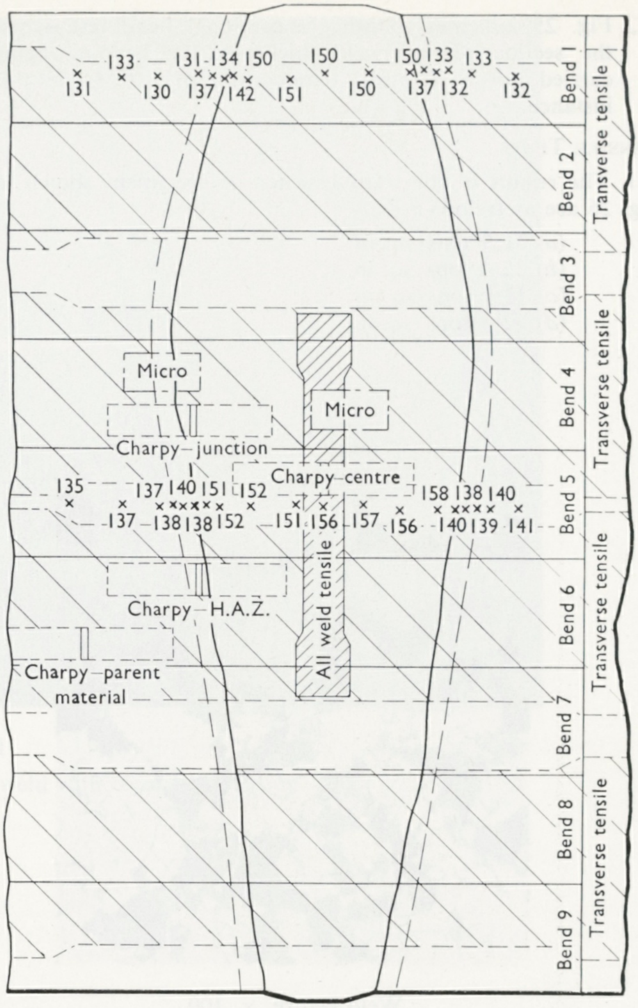


FIG. 23
13 in. electroslag weld

HARDNESS SURVEY

The VPN hardness values in the weld zone are recorded on Fig. 23.

CONCLUSION

Electroslag welding must still be regarded as a comparatively new process and development work is continuously being carried out to establish new techniques to improve quality in the "as welded" condition.

The main objective at the present time, is of course, to produce a weld structure sufficiently fine to be acceptable for "severe duty" services without normalising being a requirement.

The radiographic features of weld metal have been found superior to those of metallic arc deposits and the metal is quite free from slag inclusions and porosity, the faults mostly recurrent in weld metal. In fact, the limited defects that do occur are generally evident from the surface of the weld.

Manufacturers of heavy pressure vessels have, of course, recognised the advantages of the process, not only in quality of weld metal and increased rate of deposition but also that fabrications which would not be contemplated with thick sections using conventional welding methods can readily be welded by the electroslag process.

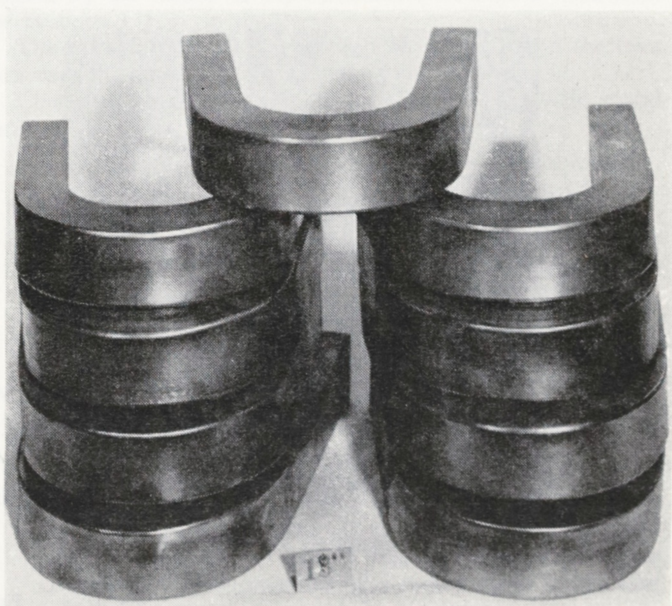


FIG. 24

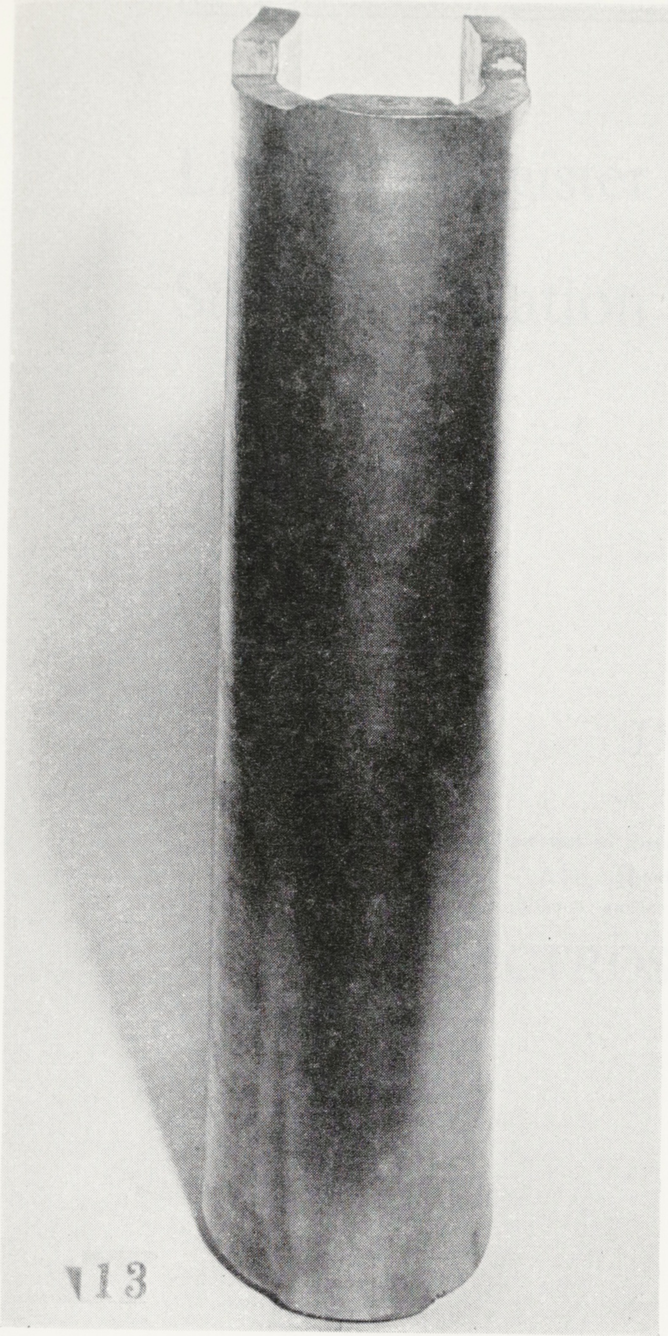


Fig. 25

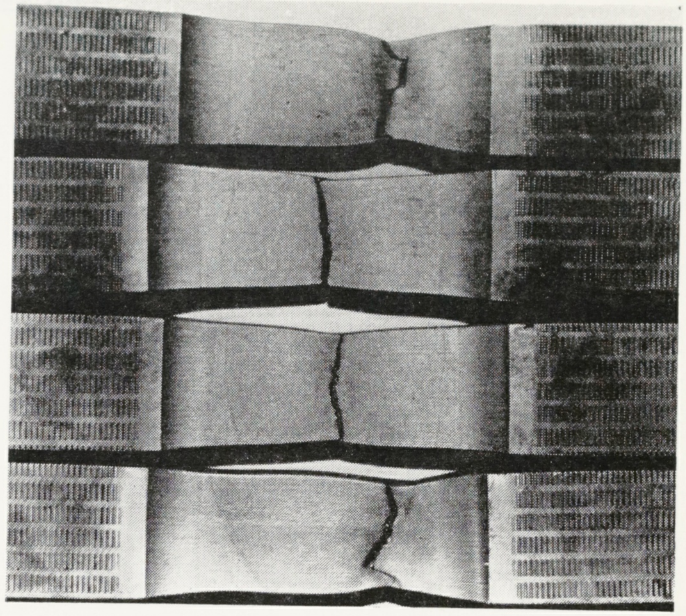


Fig. 26

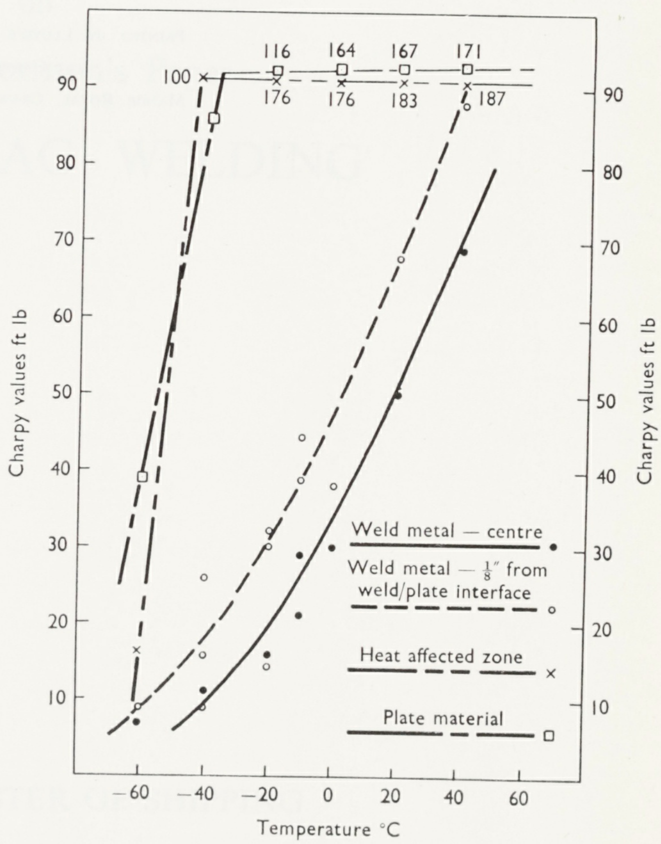


Fig. 27
13 in. electroslag weld

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Session 1963 - 64

Paper No. 1

Discussion

on

Mr. R. Morrison's Paper

ELECTROSLAG WELDING

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

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Discussion on Mr. R. Morrison's Paper

Electroslag Welding

Mr. R. E. LISMER (Crawley)

Our appreciation is due to Mr. Morrison for providing the Staff Association with a very useful paper outlining the nature of the electroslag welding process and its metallurgical characteristics. As with every new welding assembly method, the Society will have many times to consider the extent of the applications of the process and in the future, other engineering spheres will be investigating its possibilities in their particular field.

Electroslag welding was developed by the Russians in 1951, the first pressure vessel being constructed by the process in Russia in 1952. It seems that a long time has been taken in accepting its possibilities in Britain, and the delay is perhaps in some degree, due to the suspicion by Inspection Authorities that a very coarse grain structure within the deposited metal can be associated with mechanical properties detrimental to the service of the weldment. The impact values have been subject to suspect, but improved wire analysis and better operation control mean that satisfactory values can be attained even without post-heat treatment. In any case whether "as welded" or normalised it should be noted that the impact values are higher in the coarse grained region than at the centre of the deposited metal. Thus grain size is not necessarily a significant factor, and what is more important, is the segregation which may occur towards the centre line of the weld.

Mr. Morrison has indicated that there are future developments and one of these is the object of eliminating post-heat treatment. Crystals of such large size need time to grow and there are methods, analytical or mechanical, of restricting such growth to within certain limits. When such test data are examined, these show that future consideration could be given to waiving the requirement for a normalising treatment. On this, two points are worthy of consideration:—

(1) Fine grained structures contribute to good impact properties, but all coarse grained structures should not be associated with poor impact values. Welding is an example of very high purity steel making, and in the characteristics of the electroslag process, the deposits are very free from many of the gaseous and inclusion hazards. The molten weld metal is in a highly deoxidised state. There is thus much less chance of any weld failure being the consequence of a coarse-grained structure.

(2) The presence of a large heat affected zone which could affect the impact properties. Efforts have been made to reduce the extent of the zone and the problem may be compared with what is acceptable for the submerged arc weld process. High welding currents and 2-pass operation do produce some refining of the grain structure and whilst the extent of the heat affected zone is less than in the electroslag process, Charpy impact values can be just as low at some point of a narrow heat affected zone as in the wider zones.

It can be expected that of the future developments, one will concern the elimination of the complex nature of the equipment for operating the electroslag process. The "Consumable Nozzle Process" has been recently introduced to eliminate the movement of the welding head. This process can be varied in many ways to suit the requirement of the shipbuilding industry, which now has realised that all mechanised and automated processes are production necessities.

Mr. A. C. DEARDEN

First of all I would like to take the opportunity to welcome Mr. Morrison back to London where he spent some of his earlier years with the Society and where during that period we were associated in the work of plans approval. He will probably remember this period quite well, but unless he has visited the metropolis in the interval, he will, I am sure, have found that considerable changes have taken place both in the City and within the Society's offices—if he has had the opportunity to visit the latter. It would have been nice if all his associates of those days had been able to be present to hear his paper but that unfortunately is not possible. However, I am pleased to say that 40 per cent of those associates, that is—two of us—are present!

In the next place I would like to thank him on behalf of my own department in particular for the work he has put in in preparing and presenting this paper. The subject is one which concerns a welding process which is very much to the fore at present and which is rapidly increasing in use, and the paper which he has prepared giving a concise account of the leading features of the process is one which will be valuable for reference for people like myself who, although concerned with vessels incorporating this process of welding, nevertheless have limited opportunity for direct contact with it. In my own case this has extended only to the welding of plate of thicknesses well below those to which he refers in his paper.

I would now like to refer to one or two aspects of the process on which the Author may be in a position to enlarge on the information contained in the paper or perhaps to comment on. These are as follows:—

Firstly, it is noted that copper coated electrodes are used and that by reference to page 3 the copper content of the electrode is 0·30 per cent. Whilst the thickness of the copper coating is not known, I would have anticipated a slightly higher figure than this, although that quoted would appear to be sufficient to influence the characteristics of the material. However, reference to the chemical analysis given on page 10 indicates a complete absence of copper in the weld deposit and I should therefore be glad to know if this is in fact the case and what has happened to the copper in the meantime.

The reason I raise this point is that I recollect that when Mr. Bushell was Principal Surveyor for Metals to the Society he expressed a dislike of steels containing copper and I believe that some trouble was experienced with the fabrication of boiler drums in copper bearing steels in Germany at that time.

Secondly, there is the vexed question as to whether welds made by this process should be normalised on completion in all cases. It is understood, and reference to this is made in the paper on page 10, that attempts are in hand to develop a technique to produce welds having a fine grain structure in the "as welded" condition and thereby to ensure impact properties in the "as welded" condition more comparable wi

those in the plate than are indicated by the figures shown on page 12. Perhaps the Author could say something on the methods proposed.

A point which would arise where the welding of thick sections without subsequent normalising is concerned would be the effectiveness of ultrasonic examination where this method has to be adopted in place of radiography. It would be of interest to learn the technique which has been adopted for ultrasonic examination of the welds and the limit of grain size for which it can be considered effective. It is noted that the impact tests indicated on page 12 are in the normalised condition and it would be of interest to see comparable results in the stress relieved condition.

Thirdly, the question of repairs in heavy sections—say 10 in. to 14 in. thick—is of some interest and the Author's comments would be welcome on any difficulties which may be encountered in carrying out such repairs by manual welding for the case of the type of defect caused by a stoppage in the welding process. I would expect that, unless very special precautions are taken, some difficulties might be experienced in ensuring completely sound repairs in such cases. It would be interesting to know whether the Author has in fact much experience of such repairs and whether the inspection technique adopted is known to be adequate to ensure the soundness of the repairs.

Finally, I would be glad to know if the Author can state whether or not in his experience circumferential seams have yet been welded by this process on actual production work.

MR. K. O. L. NILSSON (Gothenburg)

The Author is to be congratulated on a very interesting paper on this new welding process. The reprints of the micro- and macro-structure are good and very informative.

About four years ago this type of welding was introduced in Western Europe, and in May this year Messrs. Esab, of Gothenburg, licensees of the Paton Institute, Kiev, reported the delivery of about 50 electroslag units to various countries, none of these, however, for Sweden.

The rather hesitant acceptance of the electroslag process is affected, I think, by the large grain size obtained in the "as welded" condition and subsequent low results of impact tests. This is not unique, a similar result is obtained with submerged arc welding in single run technique.

It is often said that the impact test can only be used as a comparison between two different steels with regard to their brittleness and give little indication regarding the finished welded construction as a whole. Dr. A. A. Wells shows (1) that according to wide plate tests a brittle cleavage fracture originates only in the presence of a notch, and that a low stress fracture requires in most cases a sharp notch or pre-existing crack, such as hot cracks, heat-affected zone cracks, etc. Furthermore, in case of short notches or pre-cracks the stress must be of at least yield magnitude for crack propagation to occur. Taking into account the good quality of an electroslag weld with its freedom from slag inclusion, porosity, etc., and also, when compared with conventional welds, its relative freedom from residual stresses, the chance of a brittle fracture occurring in this type of weld should be small despite its large grain size.

A Swedish firm has carried out various tests, such as N.D.T. tests (Nil-Ductility-Temperature), Wide Plate tests and also Explosion tests in order to investigate this interesting matter further. As far as can be seen from reports published (2) the results are encouraging.

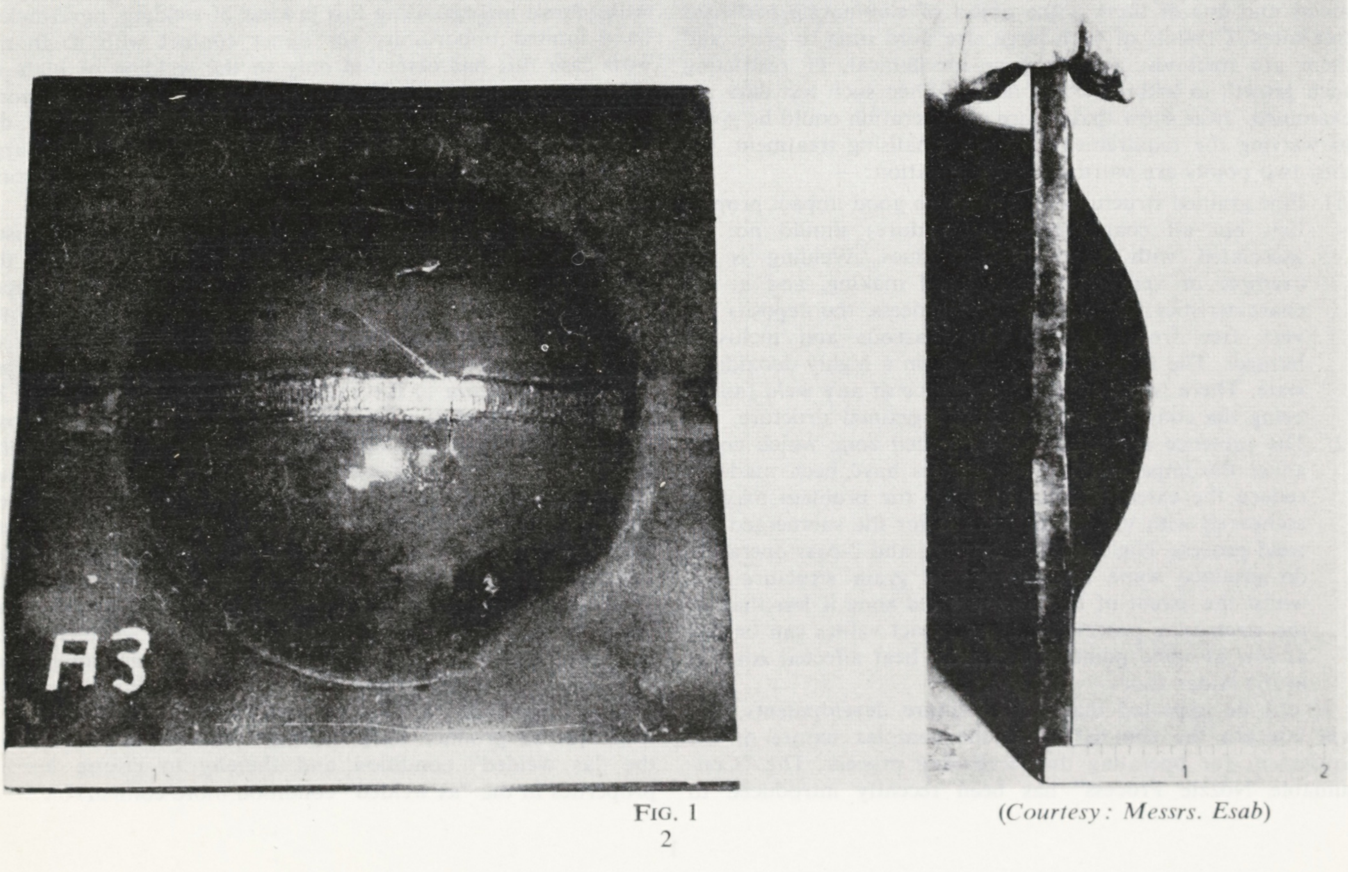

FIG. 1 (*Courtesy: Messrs. Esab*)

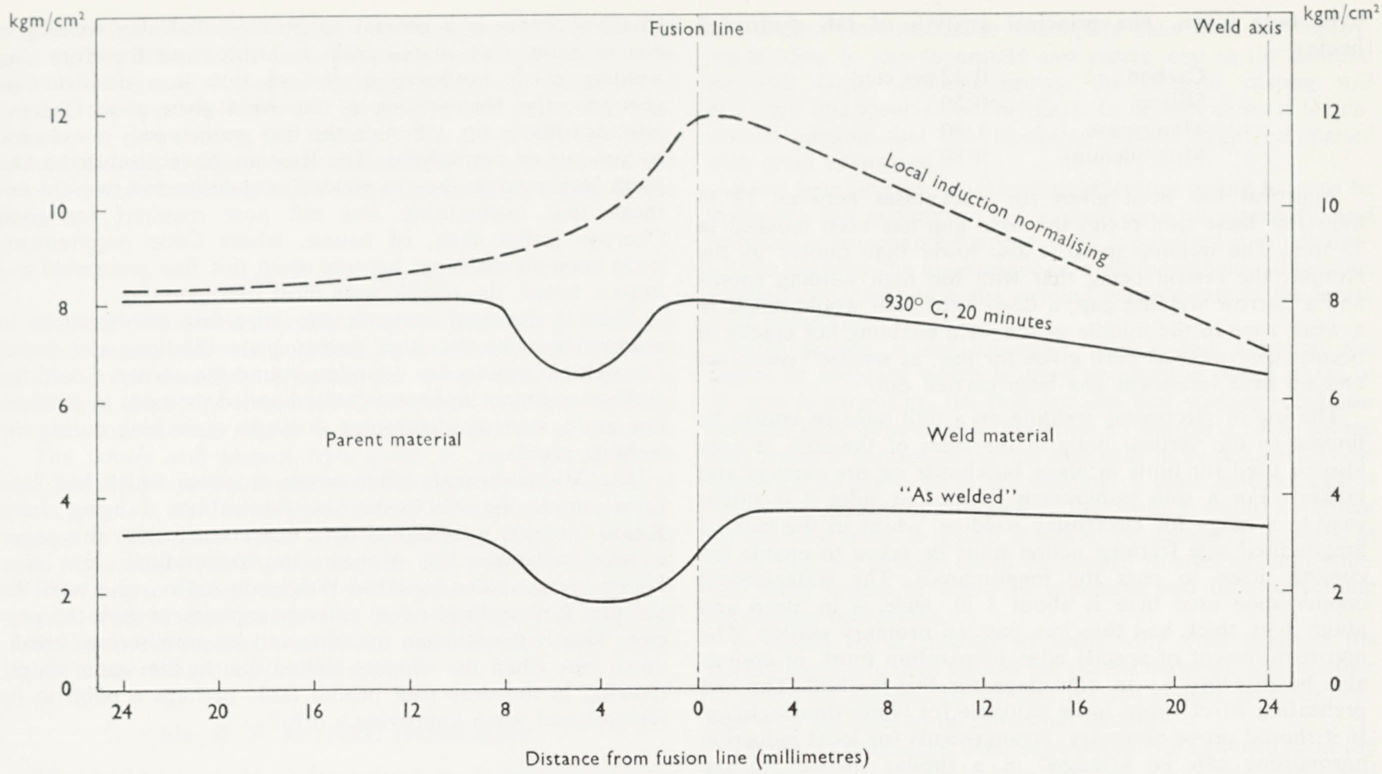

FIG. 2

Impact strength for V-notch test piece, plate thickness 40 mm., material SIS 1411, temp. $\pm$ 0° C.

Fig. 1 shows a plate, subjected to explosion test, which was carried out at Messrs. Bofors. After five shots, giving a depth of the plate of 69 mm. a fracture occurred.

As stated by the Author, development work is concentrated on improving the quality in the "as welded" condition in order to avoid the necessity for normalising. However, the firm previously mentioned has also investigated the effect on the impact strength using local induction normalising (3). In Fig. 2 some curves are shown giving the impact strength for the conditions "as welded", "normalised" and "induction normalised".

The investigation shows that good results can be obtained by local induction normalising.

For a ship surveyor it is, of course, of interest to learn that the electroslag process has been approved for welding hull structural steel and in fact also used on a ship classed with this Society. This ship was the m.s. *Laponia,* the first new-building delivered from Götaverken's new shipyard, at Arendal, where the butts between the sections are welded indoors.

The previously mentioned firm approached this Society in November, 1961, with a view to getting electroslag welding accepted for use on hull structural steel, and the requirements as per Fig. 3 were laid down by the Head Office for the approval tests.

The first tests led to the approval of OK Autrod 7+OK Flux 5 for use on grade D steel. Subsequently other approvals

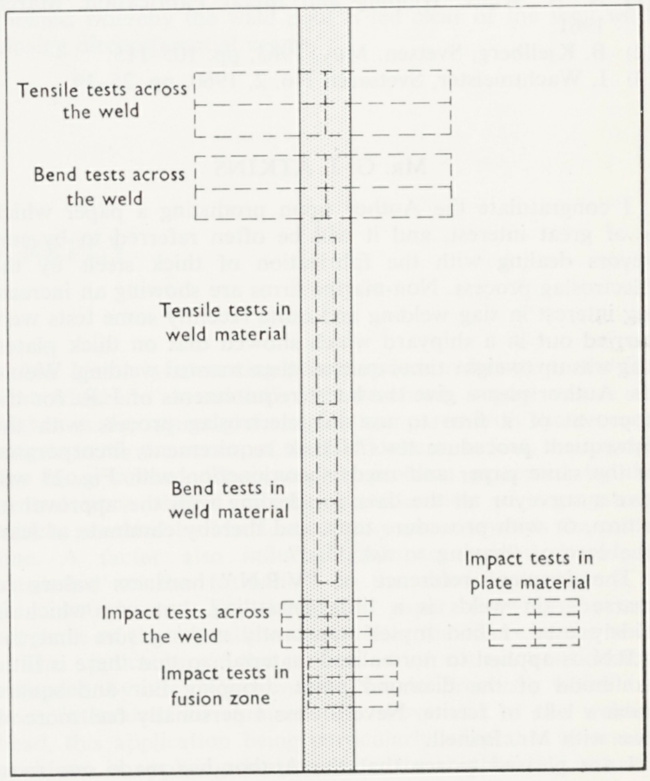

FIG. 3

Approval test for welding hull structural steel.

have been given. The principal analysis of OK Autrod 7 being:—

Carbon	0·12 per cent
Silicon	0·50 ,,
Manganese	1·80 ,,
Molybdenum	0·80 ,,

Approval has been given for thicknesses between 12–35 mm. For these thin plates the weld gap has been reduced to 15 mm. The welding speed is also lower than quoted by the Author, the reason being that with too high welding speeds and a narrow welding gap, a deep weld bath would result in a weak zone in the middle of the weld enabling hot cracks to occur. Approval has been given for the "as welded" condition and no heat treatment has been carried out.

The use of electroslag welding on a hull must of course be limited to the vertical butts in the sides of the ship, it may also be used for butts in plane bulkheads on ore carriers and tankers. For a ship transversely framed at sides it is rather easy to arrange for electroslag welding, whilst in the case of longitudinal side framing action must be taken to enable the cooling hoses to pass the longitudinals. The water-cooled copper shoe used here is about 3 in. wide, 4 in. deep and about 1 in. thick and thus can pass an ordinary scallop. The non-requirement of special edge preparation must, of course, also be regarded as an advantage for this method. The self preheating effect ought to be valuable for wintertime welding If it should prove necessary, arrangements for local induction normalising can be arranged in a similar manner to the welding itself, but so far no equipment has been constructed.

REFERENCES

(1) A. A. Wells, Welding and Metal Fabrication, March, 1961.

(2) B. Kjellberg, Svetsen, May, 1963, pp. 105–115.

(3) I. Wachtmeister, Svetsaren No. 2, 1960, pp. 25–39.

Mr. G. J. ATKINS

I congratulate the Author upon producing a paper which is of great interest, and it will be often referred to by surveyors dealing with the fabrication of thick steels by the Electroslag process. Non-marine firms are showing an increasing interest in slag welding and quite recently some tests were carried out in a shipyard which showed that on thick plates, slag was up to eight times quicker than manual welding. Would the Author please give the latest requirements of L.R. for the approval of a firm to use the electroslag process with the subsequent procedure tests? These requirements incorporated in the same paper and used in conjunction with Fig. 23 will give a surveyor all the data for dealing with the approval of a firm, or with procedure tests, and thereby eliminate at least one case of "having to ask Dad".

The frequent reference of "V.P.N." hardness values to coarse grain welds is a little unsettling, but one which is widely used. I find myself constantly making sure that the V.P.N. is applied to normalised material, so that there is little likelihood of the diamond point dropping fair and square upon a lake of ferrite. Nevertheless I personally feel more at ease with Mr. Brinell.

I am pleased to see that the Author has made one most important point. "Coarse grain can give adequate Charpy values." Conversely, fine grain does not necessarily mean good Charpys. There is a general supposition that slag welding is coarse grain, and coarse grain is brittle, and therefore slag welding needs normalising, and so that is a disadvantage economically. Normalising a slag weld gives good Charpys and also fine grain, although the fine grain is only a welcome by-product of normalising. The Russians have produced a fine grain slag weld in the "as welded" condition but this did not mean that normalising was not now required for good Charpys (other than, of course, where Code requirements insist upon normalising) because when this fine grain weld was impact tested, the results were most disappointing.

There is constant research into wire/flux combinations in association with the high parent plate dilution, and better "as welded" results are doubtless round the corner. Could the Author comment upon externally applied methods to produce fine grain, such as ultrasonics, or water quenching during the welding process.

The Middlesbrough office covers an area which has four firms employing electroslag in fabrications ranging from Kaldo furnaces to bridge girders, and consequently this paper is most welcome. Fig. 4 shows the longitudinal seam of a Kaldo being welded by Head Wrightson & Co., who were the the first firm in England to actively experiment with the process. This is the Russian machine and its most serious breakdown was when the slippers melted due to the water supply freezing in the shop roof header tank, perhaps a point to be remembered when approving a firm?

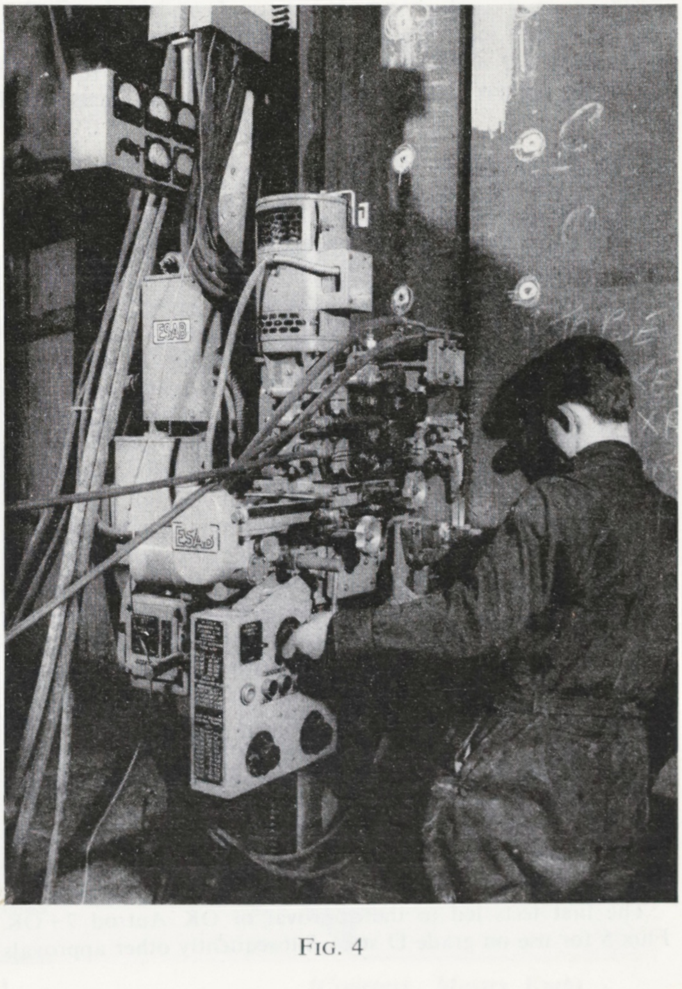

Fig. 4

Circumferential slag welding has been successfully carried out by two firms in this area. The main teething troubles were of obtaining an ultra-slow rotating roller-bed, accuracy of rolling the two sections to be bridged by the shoes, and provision for controlling the creep of the job whilst rotating. These difficulties have been successfully overcome, and we look forward to seeing much more electroslag welding in the future.

Mr. G. M. BOYD

In most discussions on electroslag welding, and particularly in this paper, there is considerable preoccupation with the coarse grain structure. I would ask what evidence there is that large grain size is in itself detrimental, and if so in what respect and in what degree is the effect observed in practice.

The tensile and impact tests given in the paper indicate satisfactory properties in the weld metal relative to the parent plate, even in the "overheated zone". These tests, however, appear to have been made after normalising. Can the Author give any comparable figures for the "as welded" condition?

Referring to the section on "weld repairs" would the Author please confirm that stress-relief heat treatment is required after any weld repairs, even if the structure had been previously stress relieved before the repair?

Mr. R. F. MUNRO (Winterthur)

The Author is to be congratulated on a timely and very well-presented paper setting forth the basic principles of a comparatively new process which is undoubtedly destined for development and more general adoption, and in my mind's eye I can see this work being referred to in many L.R. offices throughout the world for a long time to come.

Foundries engaged in the production of large steel castings are tending more and more to simplify their work by employing welding to join separately cast pieces, and as the demand for still larger castings grows, the integral casting will probably disappear from this field. In certain cases it is considered possible that the electroslag process might be applied with great advantage.

Some handling and positioning difficulties would have to be overcome but economics are a great spur to ingenuity, and provided a reasonable number "off" is required it is not difficult to visualise the appearance of multi-positional, adjustable, powered welding jigs in the welding departments of the larger steel foundries.

Sand casting is not an exact science and although the degree of accuracy which is often obtained in pieces of up to 20 tons is surprising, the fact remains that surface roughness coupled with difference in thickness of the parts to be joined may make it impossible for the slag bath to be retained. These problems do not arise in the welding of plate material and it would be greatly appreciated if the Author would comment on the tolerances to which it would be necessary to work in order that the process could be applied to steel castings without being forced to resort to expensive and time-consuming machining operations.

It would be of interest to learn with what pressure the copper shoes are held against the parent material, and whether it would be possible to arrange for the copper faces in contact with the steel to be somewhat flexible.

Have any difficulties been experienced with porosities in the weld metal due to air leaking between the copper shoes and the parent metal surfaces?

Finally, I would like to suggest to the Author that the section on "Boiler Drums and Pressure Vessels" would be greatly enhanced by the addition of a description of the method whereby the weld pool is led clear of the shell when closing circumferential seams.

AUTHOR'S REPLY

The Author wishes to thank the members who have taken part in the discussion or contributed by writing, and before dealing with those interesting points, attention is drawn to the erratum contained in the publication.

Page 10, Fig. 14 shows the macrostructure of a longitudinal section of an electroslag weld before normalising and this photograph should be inverted to be in accordance with the upward direction of welding of the process. Solidification commences at the weld boundaries and the dendrites grow in a direction opposite to that of the temperature gradient, the dendrites in the central zone then "point" in the direction of welding.

To Mr. Lismer

A retrospect establishes that the process was first thought of by a Russian in 1888 and was possibly patented at that time, the process being subsequently developed for production in 1950 at the Paton Welding Institute of Kiev.

Mr. Lismer's interesting points include reference to the fact that whether "as welded" or normalised, impact values are higher in the coarse grain region compared with those occurring towards the centre of the weld where, in fact, the crystals are more refined. The complex problem presented by this central zone should include consideration of segregations which, as stated by Mr. Lismer, may occur towards the centre line. A factor also influencing impact values should be attributed to the formation of the dendritic structure in this central zone, Fig. 14 in the paper, showing the convergence of the crystalline structure in the centre of the weld.

The application of the process is, of course, considerably extended by the "Consumable Nozzle Process", which has been introduced to eliminate the movement of the welding head, this application being particularly adaptable to resolve problems in applying the process in the shipbuilding industry where new and improved developments in production methods are always keenly sought.

To Mr. Dearden

Mr. Dearden's kind words of welcome are indeed a pleasant reminder of my earlier marine and non-marine service in London.

It is a fair assessment to say that the copper coating on electrode wires has not been the cause of faults in weld metal but this is due to the careful checking of weld metal analyses to ensure that safety limits are not exceeded. The composition of electroslag weld metal is substantially influenced by the parent plate analysis due to the effect of dilution and this can be observed from Fig. 20 in the paper. It is, therefore, clear that the copper content the weld metal receives from the electrode coating must be less than that resulting from submerged arc welding where there is less dilution from the parent material, provided, of course, the same size of electrode wire is used. Electroslag welding with the technique described in the paper would have a copper content in the weld of about 0·15 per cent to 0·25 per cent which is not considered as significant. Copper from the coating of electrode wire enters into solution with iron during welding and with these percentages there is not the undesirable intergranular effect that would occur if copper was introduced as a separate element or concentrated at a point during welding. It may be of interest to note that with welding processes employing a high heat density and a relatively small size of wire, for example the CO_2 process, the percentage of copper due to the coating of the wire may reach proportions that do require consideration. As there does not appear to be a more suitable metal on the market for the purpose of wire preservation than copper, electrode wire of very small diameter may require to be supplied without a coating and be protected from corrosion by silica gel sealed in the container.

It is known that attempts to produce fine grain structures in the "as welded" condition have included such expedients as the additions of certain elements, viz.: aluminium, vanadium and others to the weld pool. Measures of refinement have also been obtained by subjecting the weld pool to vibration by mechanical or ultrasonic means.

However, refinement of the weld structure only does not mean that the problems of the weld joint as a whole are overcome. These methods do not alter the thermal conditions that produce the heat affected zone characteristics, therefore problems associated with these zones will not be eliminated without a suitable heat treatment.

The basic reason for the coarse grain structure, as stated in the paper, is the slow rate of cooling, therefore, in theory, if a welding technique is established to increase the cooling rate such as would occur by welding faster, then this would control grain size and minimise the extent of the heat affected zones. However, achieving this in practice and also matching the high standard of physical properties of the normalised weld is not a simple matter.

The methods of ultrasonic examination used to examine the welds of production components are briefly referred to on page 9 of the paper but it would not be feasible to specify quantitatively the limit of grain size affecting accurate interpretation. This difficulty constitutes a challenge to industry and there are now claims of successsful ultrasonic methods having been developed for this purpose.

It is known that the results of impact tests from electroslag welds subjected to stress relieving only, can be satisfactory in mild steel but less satisfactory in low alloy ferritic steels, particularly in their heat affected zones.

The results of impact tests from mild steel electroslag welds are improved by stress relieving but remain lower than those in the normalised condition. Reference, therefore, could be made to the results in Fig. 5. It is perhaps relevant to mention here, that where normalised electroslag welds undergo a stress relieving operation, due to additional assemblies incorporating conventional welding, such as submerged arc circumferential seams or nozzle attachments, the reflection in impact testing is a marginal improvement together with a reduction in the scatter of values as compared with those in the normalised condition.

The most practical method of rectifying a serious defect in an electroslag weld of a thick component is to machine out the completed or partly completed seam and re-weld. With welds of 5 in. or 6 in. thick the procedure for the repair of a fault is the same irrespective of the welding process. Although the submerged arc weld may contain a smaller fault, it will be appreciated that if this occurs at mid-thickness the actual excavation to remove the defect can be extensive. The walls of the excavation should be dressed smooth and be angled to facilitate side fusion during repairs which are carried out manually. The soundness of repairs can be satisfactorily examined either radiographically or ultrasonically, the method of non-destructive testing specified for contract being, of course, applied to the repair.

There are a number of instances in the U.K. where circumferential seams have been completed with electroslag welding but no case has come to my attention where these seams have occurred in severe-duty pressure vessels requiring a subsequent normalising although literature from Russia on the subject of large boiler drums indicates the successful incorporation of such circumferential seams.

To Mr. Nilsson

Mr. Nilsson's contribution indicates the favourable results of metallurgical research on electroslag welding and his account of the process being applied to the welding of hulls is of particular interest.

The impact results from weld metal should always be examined in conjunction with the impact values of the material being welded.

Some results of tests carried out by Messrs. Babcock & Wilcox Ltd., Renfrew, on Charpy "V" notch specimens tested at $+20°$ C. and relevant to mild steel boiler quality plate with electroslag welds in the "as welded" and the normalised conditions are shown in Fig. 5, where it will be noted that the region of lower impact values in the heat affected zone of the "as welded" test plate is eliminated by normalising. A similar change would be anticipated in this region shown in Fig. 2 if the duration of the normalising treatment on the 40 mm. thick plate was the accepted standard of one hour per inch of plate thickness instead of the 20 minutes stated thereon. The impact results would then be expected to conform more closely to those from the test plate normalised by the local induction method also shown in Fig. 2.

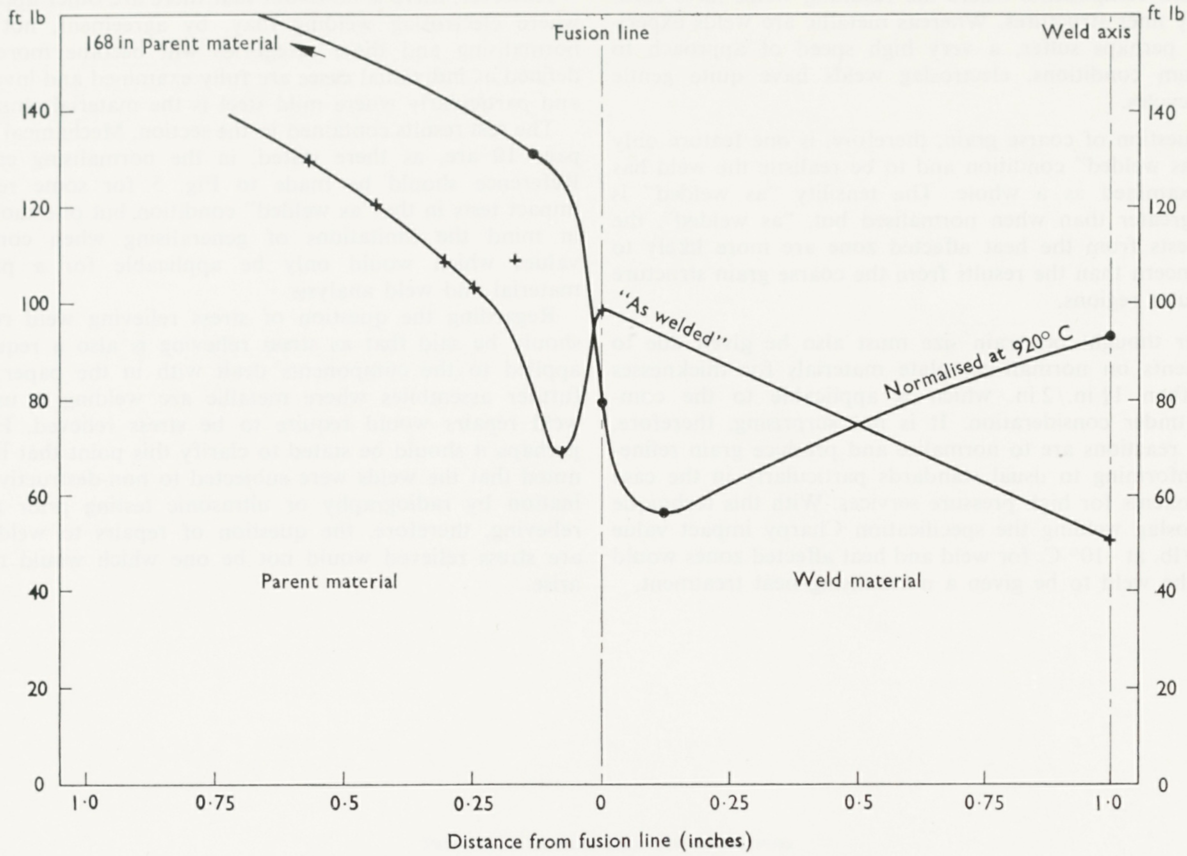

FIG. 5

Charpy V-notch strength on mild steel boiler quality plate at +20° C.

TO MR. MUNRO

A lightly dressed plate surface in way of the sliding shoes has proved to be an adequate standard for trouble-free welding. The surface condition, however, of 14 in. thick forged billet material does contain some quite deep irregularities but dressing these areas into smooth undulations is found to be sufficient. In all cases a trial run of the machine to test the shoe assembly along the seam is necessary prior to welding.

The surface of steel castings could be dressed in a similar fashion or a technique improvised where irregularities on the surface could be smoothed by applying a suitable heat resistant compound.

The shoes are held to the weldment with a load of about 10 lb. and, arranging the point of application near the top of the shoe, that is the leading edge, assists the shoe to manœuvre surface irregularities.

The question of air leakage into the weld metal from the shoe faces is not a valid one as this air is free to expand outwards. This point, however, is relevant to the starting blocks which should be tack welded to allow entrapped air to expand outwards, otherwise such large volumes if forced into the weld, will produce piping. Slides illustrating this feature were included in the presentation of the paper.

The closing of circumferential welds is achieved by securing the inner shoe and leading the outer shoe along side plates to contain the weld. The triangularly shaped surplus metal is subsequently removed and the area dressed to the contour of the outer surface of the shell. Again, the presentation of the paper did, in fact, include slides showing the closing of an actual circumferential weld.

TO MR. ATKINS

With the wide range of components, specifications and perhaps welding techniques, the procedure of consulting Headquarters is desirable at present, as, rather than adhere to a rigid set of tests or conditions, the flexibility of individual consideration is infinitely preferable.

Mr. Atkins raises the question of methods applied externally to produce controlled grain structures in the "as welded" condition and reference should be made to the comments in reply to Mr. Dearden's question which, in this respect, is similar. I believe the "break through" will result from a study of the solidification aspects of the weld rather than from externally applied methods. Water quenching would not appear to be a suitable method as moisture contamination of the process would be most undesirable.

TO MR. BOYD

A study of the solidification of the weld metal deposited by the process of electroslag welding is, of course, essential to an appreciation of the subject. This feature differs from that of conventional welding processes applied to the construction

of pressure components where the resulting welds have comparatively finer structures. Whereas metallic arc welds experience, or perhaps suffer, a very high speed of approach to equilibrium conditions, electroslag welds have quite gentle thermal cycles.

The question of coarse grain, therefore, is one feature only of the "as welded" condition and to be realistic the weld has to be examined as a whole. The tensility "as welded" is slightly greater than when normalised but, "as welded", the impact tests from the heat affected zone are more likely to cause concern than the results from the coarse grain structure of the outer regions.

Further thought on grain size must also be given due to requirements on normalising plate materials for thicknesses greater than $1\frac{3}{4}$ in./2 in. which is applicable to the components under consideration. It is not surprising, therefore, that first reactions are to normalise and produce grain refinement conforming to usual standards particularly in the case of components for high pressure services. With this technique of electroslag welding the specification Charpy impact value of 20 ft./lb. at –10° C. for weld and heat affected zones would require the weld to be given a normalising heat treatment.

However, there is no doubt that there are other applications where electroslag welding may, by agreement, not require normalising and these categories will become more clearly defined as individual cases are fully examined and investigated and particularly where mild steel is the material employed.

The test results contained in the section, Mechanical Testing, page 10 are, as there stated, in the normalising condition. Reference should be made to Fig. 5 for some results of impact tests in the "as welded" condition but one should bear in mind the limitations of generalising when considering values which would only be applicable for a particular material and weld analysis.

Regarding the question of stress relieving weld repairs it should be said that as stress relieving is also a requirement applied to the components dealt with in the paper, due to further assemblies where metallic arc welding is used, any weld repairs would require to be stress relieved. However, perhaps it should be stated to clarify this point that it will be noted that the welds were subjected to non-destructive examination by radiography or ultrasonic testing prior to stress relieving, therefore, the question of repairs to welds which are stress relieved would not be one which would normally arise.

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LONGITUDINAL STRESSES IN MODERN CANADIAN GREAT LAKES BULK FREIGHTERS

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LONGITUDINAL STRESSES IN MODERN CANADIAN GREAT LAKES BULK FREIGHTERS

By F. S. J. McKINLAY

The number of published investigations on longitudinal stresses induced in large Great Lakes vessels is small and additional information on the subject is desirable. The modern Canadian-owned and built Great Lakes bulk freighter of maximum Seaway size is likely to be the principal replacement vessel in Canada for some years to come and an extensive examination of the great variety of loaded and ballasted conditions of which she is capable during the operational season would be of benefit to the industry. This paper, presented in three parts, is intended to satisfy this need.

The first part deals with the historical evolution, environmental restrictions and functional requirements of this type of ship.

In the second part of the paper the Society's method which greatly reduces the time required to compute longitudinal bending moments is modified for specific use in assessing stresses in large lakers. Permissible longitudinal stresses are determined for lake ships between 400 ft. and 1,000 ft. in length.

The final part investigates the longitudinal bending moments and stresses in a maximum-sized Canadian bulk freighter, induced by the loaded and ballasted conditions peculiar to her service. Realistic and actual recorded distributions of cargo and ballast are considered in association with practical draughts, trims and service experience.

The paper supports the premise that these large ships are operated and handled with competency and that the present design is unsurpassed in the shipping industry for the basic function of transporting large bulk cargoes economically.

PART I—GENERAL

Introduction

The River St. Lawrence has many rapids and shoals prohibiting commercial navigation between Montreal and Prescott, and until recently six small Canadian canals were in use having a combined lift of 210 ft. involving 21 lockages. These canals limited ships to 255 ft. overall by 44 ft. in breadth and 14 ft. maximum draught. Therefore before the St. Lawrence Seaway was opened in the spring of 1959 all freighters operating on the lake system drained by the River St. Lawrence could be broadly classified into one of two types depending on whether their length was greater or less than 255 ft. The former is known generally as upper lakers and the latter, the no less industrious family, known as canallers. The canallers, which on the whole were Canadian owned, reigned supreme below Prescott and infiltrated very successfully into

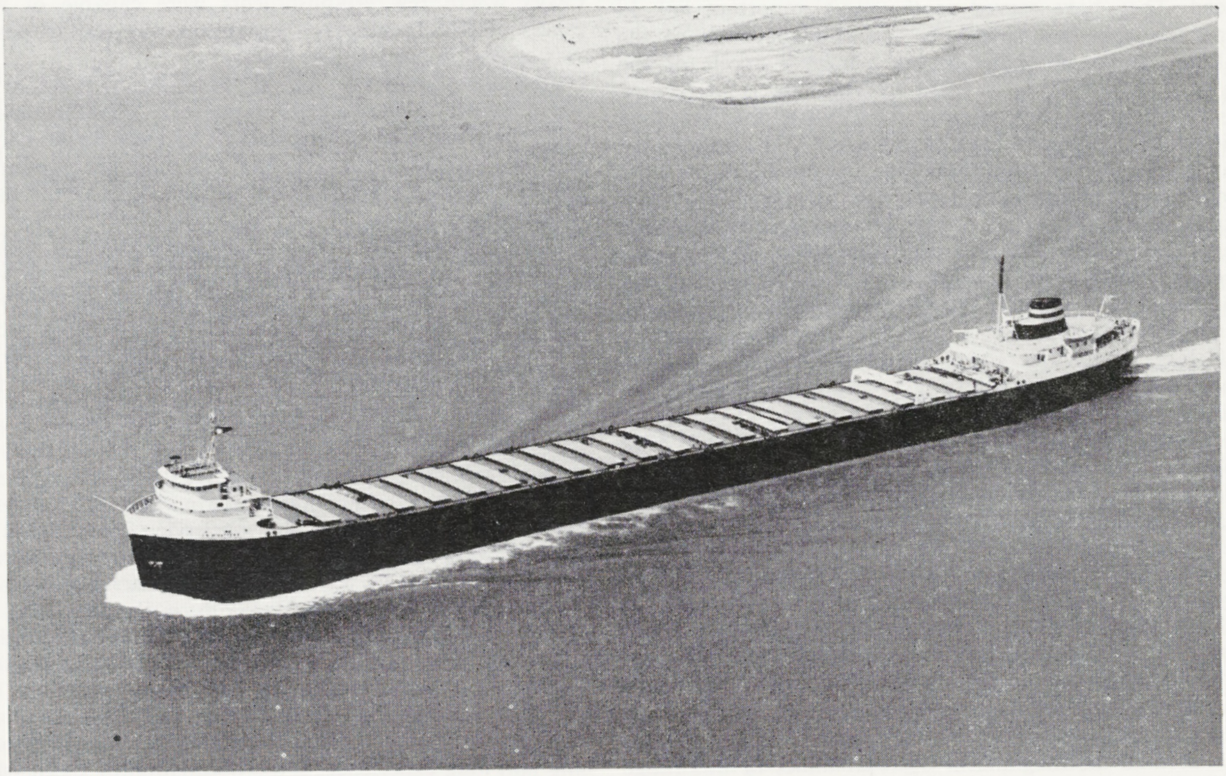


Plate 1

J. N. McWatters on the River St. Lawrence

the realm of the upper lakers which on the other hand were condemned to spend their working lives confined to the 95,000 square miles of the fresh water lakes of North America.

The St. Lawrence section of the Seaway provided the larger ships with access to the lower reaches of the river and distinctly upset the balance of power between the two groups. Experience of operating large vessels on the lakes proved the economical advantage of the bigger ships and the monopoly of the picturesque, work-scarred canaller was usurped. Some canallers continue to operate and indeed a few have been built and completed as specialised carriers particularly in the pulpwood and paper trades since the Seaway project commenced but as special surveys became due on the older ships their numbers were greatly reduced. It was the ending of an era with many having fulfilled their destiny, some locally at the shipbreakers and some, ironically enough, had their last trip across the Pacific or Atlantic to meet the same fate. Some, again, live a few more borrowed years as drilling barges and in similar lowly services.

The character of the river has changed now that the large upper lakers have become the major replacement tonnage ; the impetus imparted by the Seaway and the consequences of new environment have created reassessments and readjustments in the design of this unique type of ship.

Evolution of Large Lakers

The Great Lakes were assigned an imperial position in the development of North America. Stretching, with their St. Lawrence outlet, half across the continent, they enabled the raw materials of this vast territory to be united in such volume and at such cost that older industrial nations were easily outdistanced. They have served as the most important link in bringing the abundant foodstuffs and raw materials of the central and northern plains to the hungry markets of the world.

This sociological development kept alive the maritime traditions of the heterogeneous people while they consolidated as a nation which as a whole was able to concentrate on their new-found manufacturing and agricultural resources. Above the rapids of the River St. Lawrence, lake ship operators and builders and makers of marine equipment and supplies worked in security from competition to build up their shipping industry. Under such conditions the birch bark canoe became the bulk freighter.

Early large upper lakers were essentially American in concept and design having their origin in the discovery of ore deposits in the Lake Superior region well over 100 years ago. The first recorded shipment of ore from this area was in 1854 and the first iron bulk freighter, the *Onoko,* was built in Cleveland, Ohio, in 1882 (Ref. 1). She was 302 ft. in length and 3,000 tons deadweight.

The discovery of the vast Mesabi Range of deposits at the western end of Lake Superior in 1890 was followed by the construction of the Poe Lock at Sault Ste. Marie in 1896. This lock linking Lake Superior with Lakes Huron and Michigan was 800 ft. by 100 ft. with an average depth of 20 ft. and made larger ships possible. The first shipment of ore from this range was made in 1892 when a mere 4,245 tons were handled and the next year the figure had increased to 613,620 tons. Since that time the magnificent open-pit mines of this range have consistently contributed about three-quarters of all the ore going down from Lake Superior (Ref. 2) and more than half of the total mined in the whole United States. It was inevitable that this period of rapid industrial expansion in the last decade of the 19th century should set the stage for the appearance of the true large upper laker. Five-hundred feet in length was reached by the turn of the century.

The first grain elevator on the lakes was built at Buffalo in 1842 and although the carriage of grain pre-dated ore and continued throughout to be an important commercial commodity, ship design advanced mainly in the ore traffic with the grain carriers subsequently assimilating proven advantages.

Until about 1904 structural arrangements of iron and steel lake ships were "sea-going" in concept having 'tween deck beams and hold pillaring. In this year, however, a ship called the *Augustus B. Wolvin* was specially designed for the ore trade and was probably the prototype of the present-day large Great Lakes bulk freighter (Ref. 3). She is reputed to be the first vessel having deep side tanks and transverse deck arches. Her moulded dimensions were 540 ft. × 56 ft. × 32 ft. and overall length 560 ft. which was the largest up to that time. This distinction to fame was short lived, however, as two years later the first 600-footers appeared.

For about the next 50 years maximum dimensions were to vary little. During periods of trade recession over these years the smaller ships were always laid up first proving conclusively that large ships are favoured but until the St. Lawrence Seaway proposal began to take tangible form many operators considered the 620-footers of about 33 ft. to 35 ft. deep to be the convenient economical limit of size (Ref. 4) having regard to the difficulties of manœuvring in restricted waters, the effect of restricted draughts and beams at loading terminals and to existing docking or slipping facilities. In addition, the maximum length was limited by existing shipyards on the Great Lakes where few establishments were capable of much in excess of 620 ft. although expansion was readily possible as was proved in the last five years.

Coincident with the Seaway becoming a material fact in 1959, the economical length of the laker increased, metaphorically overnight, to 730 ft. overall and the Seaway authority who had intended that 715 ft. was long enough for safety, granted special dispensation to accommodate this new development.

One significant advance in structural design was incorporated in the Canadian owned *Lemoyne* built at Midland, Ontario, in 1926. She was the first upper laker to have longitudinal side tank bulkheads carried up to the spar deck.

It is traditional in upper lakers that the freeboard or weather deck is named the spar deck and the top of the wing ballast tanks is called the main deck ; this convention will be adhered to throughout.

Service Area

For some part of every day of service on the lakes land is in sight, every day the vessel passes over areas having less than 30 ft. depth of water, for large stretches the navigation channel is less than 500 ft. in width, in Seaway locks there is 4 ft. 6 in. total clearance in beam and the ships under consideration are longer overall by 80 ft. than the 650-ft. *Empress of Canada.*

Navigation on the lakes, based primarily on a rigidly disciplined ritual, customarily learned by patient apprenticeship and long experience, is the occupation of specialists some of whom may not be over-endowed with academical qualifications but the practical proof of their ability is displayed on every trip. Many have known no other working life but afloat

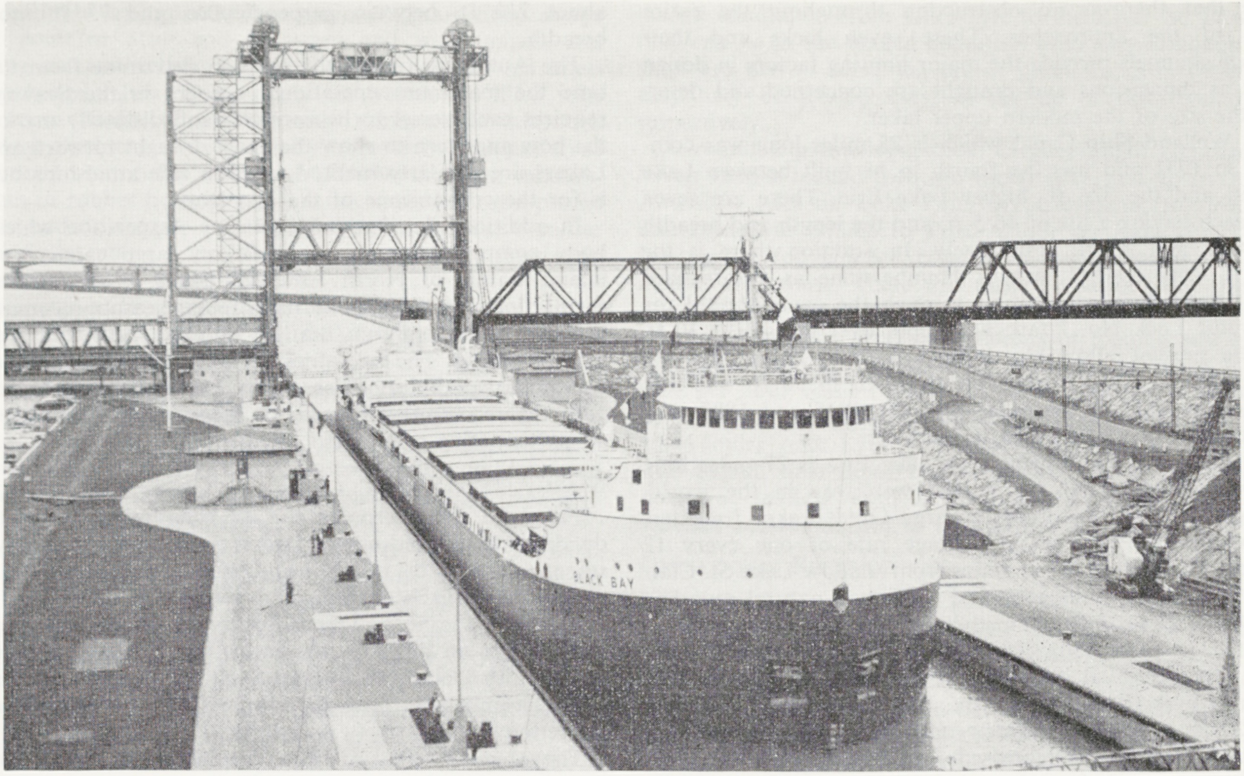


PLATE 2

Black Bay entering St. Lambert Lock, Montreal

on the lakes and in the operation and handling of their vessels they are a living part of her. This prominent feature explains to a large extent why mishap and even disaster is so often avoided in the face of undesirable characteristics embodied in upper lakers. Only below Montreal is pilotage compulsory for the large laker.

The five Great Lakes of North America whose combined area exceeds that of England, Scotland and Wales, form the home territory of the upper laker and are linked, chain fashion, by canal and navigable waterway to drain finally into the River St. Lawrence at the eastern end of Lake Ontario. The lakes in themselves do not present restrictive problems to the naval architect or marine engineer but four natural barriers between Montreal and Lake Superior require man-made development and handicap shipping by putting a restraining hand on free design. Proceeding westwards from Montreal these barriers are as follows:—

1. The River St. Lawrence shoals and rapids between Montreal and Prescott, now overcome by the Seaway canals and channels.
2. The Niagara River and Falls between Lake Ontario and Lake Erie, completely by-passed by the Welland Ship Canal.
3. The Detroit River, Lake St. Clair and St. Clair River connecting Lake Erie and Lake Huron, made navigable by dredging.
4. St. Mary's River and St. Mary's Falls between Lake Huron and Lake Superior, successfully overcome by dredging and by the United States and Canadian Locks at Sault Ste. Marie.

Longitude 63° W. on the north shore of the River St. Lawrence is the eastern legal limit of Lakes and Rivers service but the young town of Seven Islands which is the southern outlet for the new and expanding ore fields of Labrador and Eastern Quebec Province provides the practical terminus. Here the water is salt and the long low swell coming in from the Atlantic between Newfoundland and Cape Breton Island across the Gulf of St. Lawrence is still in evidence. Some 30 miles below Quebec City salinity disappears and about midway between Quebec and Montreal tidewater ceases.

Navigationally, the 900 miles of the River St. Lawrence can be divided into three main sections. Below Montreal, for over 700 miles to the mouth of the river, the channel is never less than 35 ft. deep which accommodates large ocean-going ships while above Prescott for some 60 miles to Lake Ontario the channel is in excess of 27 ft. deep and between Montreal and Prescott lies the 126 miles of Seaway locks, approach channels and shallow lakes.

There are seven locks (five Canadian and two American) in the St. Lawrence portion of the Seaway and the aggregate lift is about 210 ft. depending on the state of the river. The length of each lock from the breast wall to the fender is 766 ft. and the width of each chamber is 80 ft. with a depth of 30 ft. over the sills of the lock gates. The limiting depth of water is 27 ft. in way of some of the approach channels to the locks and the maximum permitted draught is at present 25 ft.* The draught is expected to be increased to 25 ft. 6 in. in the near future when all the channels have been dredged and checked to

* Increased to 25 ft. 6 in., April 1963.

ensure that there is no obstruction throughout the entire length of the approaches. These seven locks and their entrance channels provide the major limiting factors in design so far as dimensions and draught are concerned and determine the size of the modern upper laker.

The Welland Ship Canal which is 25 miles long was completed in 1931 and was the fourth to be built between Lake Ontario and the 326 ft. higher Lake Erie. There are seven locks each having a lift of 46·5 ft. and the length and breadth is 800 ft. and 80 ft. respectively. In addition there is the 1,380 ft. long control lock at Humberstone as a safeguard against Lake Erie overflowing through the canal should the uppermost Lock No. 7 fail. The depth over the sills is 30 ft. and the present allowable draught is 25 ft. 6 in. but the reaches between locks could be deepened to 30 ft. The first three canals built in this area in 1829, 1845 and 1887 only permitted passage of the 255 ft. canallers.

The Detroit River flowing into Lake Erie is 27 miles long and is reputed to be the busiest waterway in the world. During the navigational season large Great Lakes freighters pass Windmill Point at an average rate of one every 12 minutes. The Detroit River flows from shallow Lake St. Clair which is about 25 miles long and its deepest natural sounding is 20 ft. A 27-ft. deep channel is dredged across the lake to meet the 34 miles long River St. Clair flowing from Lake Huron. By continual dredging the allowable draught of this stretch of over 80 miles of rivers and lake has been increased from 20 ft. to the present new maximum of 25 ft. and this will possibly be further increased. Actual depth of water available fluctuates and depends entirely on nature for supply.

St. Mary's River connects Lake Huron with Lake Superior, the largest expanse of fresh water in the world, and is 25 miles long with a new maximum allowable draught of 25 ft. Here also dredging is proceeding but the final say in permissible depth is held by nature. Winter snowfall in the Superior catchment area, deforestation and run off in the spring all contribute to the uncertainty of lake levels and to dependent draught.

Near the head of St. Mary's River is the Rapids Section at Sault Ste. Marie where there are five locks in parallel and one of these on the Canadian side is too small for large lakers. Three of the remaining four locks on the American side can each accommodate maximum sized lakers while the fourth American lock, the Poe, is at present closed to navigation and is being enlarged to 1,000 ft. long by 100 ft. wide by 32 ft. in depth. The new lock will be in service in 1965 and will release dimensional restrictions for lakers operating above the Welland Canal but it is unlikely that Canadian owners will be immediately influenced by this extension to size since their sphere of operation includes Lake Ontario and the River St. Lawrence.

Dimensional Restrictions

The River St. Lawrence canals and approaches, the Welland Ship Canal between Port Weller and Port Colborne and the Canadian Lock at the St. Mary's Rapids Section at Sault Ste. Marie come under the jurisdiction of the St. Lawrence Seaway Authority and are conjointly termed the Seaway. In the two former sections the Authority permits vessels not exceeding 715 ft. O.A. and 72 ft. beam to transit the Seaway but subject to special instructions vessels not exceeding 730 ft. overall length and 75 ft. moulded or 75 ft. 6 in. overall beam may also transit the Seaway in the navigation season and in practice these restrictions have produced designs which are about 710 ft. between perpendiculars and 75 ft. moulded breadth.

The Authority retains the right to determine from time to time the maximum operational draught in the Seaway and requires each vessel to be correctly and distinctly marked on the bow and stern to show the exact draught forward and aft. Lakers are similarly marked on each side amidships but this is for the convenience of the operators.

In addition, the Dominion Marine Association which is a body composed mainly of Canadian cargo carriers on the inland waters of North America, issues to its members at regular intervals throughout the shipping season, recommendations for the maximum draught at other restricted areas of the waterways. The operators do not know how much actual water is available at these points and endeavour to comply with the recommendations since these are in the interests of all participating parties.

Navigational Season

For load line purposes the 12 months of the year are divided into the midsummer, summer, intermediate and winter seasons denoted by MS, S, I and W respectively and the dates during which each applies are given in Fig. 1.

In winter the St. Lawrence Section of the Seaway is closed at midnight on the 30th November, the Welland Ship Canal on the 15th December and the Sault Ste. Marie Canal on the 12th December but these dates are subject to alteration at the discretion of the Seaway Authority.

The premiums that Underwriters charge for the insurance of vessels operating later than these dates are increased steeply due to the additional hazards of winter navigation. Because of the severe ice conditions, navigation through the canals and locks is impossible during the winter months from about mid-December through mid-April. Therefore, the vast majority of great lakes vessels are laid up during this period although a very few do operate for the entire winter period on specialised services which involve navigation only on one lake.

The navigational season, therefore, lasts for approximately eight months of each year and of this the midsummer season amounts to $50\frac{1}{2}$ per cent, summer to 12 per cent, intermediate to 19 per cent and winter to $18\frac{1}{2}$ per cent.

Winter Lay Up

Although some ships operate during the first two weeks of December, the lay up period is normally from the end of November until the middle of April during which time temperatures may be as low as minus 30° F. with sub-zero temperatures being experienced for many days on end. Detailed and precise precautions must be taken if the ship is to emerge in the spring in pristine condition. Before the crew leaves at the end of the operational season all water must be drained from the hull, main and auxiliary engines, steam and water piping, winches and windlass, and heating, drinking and sanitary services throughout accommodation spaces. In addition the sterntube, sea inlets, scuppers and discharges are protected.

The vessel may be in the light condition or loaded with storage grain. In either case, for efficient drainage, she must trim by the stern even to the extent of removing storage grain from the forward hold taking into account her final trim when the boilers are dry. At her winter berth she is securely moored with numerous long leads and anchors are usually run out.

To facilitate lay up procedure drain plugs are fitted at the lowest point or after end of steam and water pipes, and steaming out connections are fitted to the body of sea inlets and discharge valves. These inlets and valves are filled with hot viscous black oil mixed with some tallow although individual chief engineers generally have their own preferred concoctions; sterntubes are also filled with the same mixture. A length of rubber hose, plugged at the upper end, and placed in open-ended scupper pipes projecting above the weather deck and protruding out of the shell prevents fractures should these exposed pipes freeze up.

Triple expansion engines with Scotch boilers are almost universal in the older vessels and coal is the principal fuel but a few have been converted to burn oil. Since World War II diesel engines and geared turbines with oil fired watertube boilers have been struggling one against the other for supremacy as main propulsion units with each adherent confident they have the better engine. Each type, however, requires different but equally meticulous attention paid to it as do the associated auxiliaries.

Machinery and auxiliaries are opened up for drainage, internal examination and thorough overhaul and some of the larger ships now maintain a temperature of 48° F. to 50° F. in the engine room throughout the lay up period. Deck machinery and particularly the steering engine and gear come under careful scrutiny. Time spent now in servicing essential equipment may save valuable days and even weeks during the hustle and bustle of the all too short operational season.

All shell valves are securely closed and made watertight as are all openings on the weather deck and topsides. In effect, the two principal objects of the precautions against frost damage are to ensure beyond doubt that all water is removed from inside the ship and that entry of outside water is impossible.

For a rather different reason all movable exposed gear and fittings such as lifeboat stores and equipment, boat falls, navigation lights, etc., are removed and stowed under lock and key. A shipkeeper watches over the vessel during this period, taking soundings throughout the ship at regular intervals and a temporary self-powered pump is sometimes fitted in the engine room for emergency purposes.

The total work entailed in the smaller ships generally occupies the whole engine room staff for two or three weeks after the close of navigation and the last man might manage home for Christmas. In the newer larger turbine-driven ships "winterising" may not be completed until the middle of January.

Typical Round Trip

Simplicity in handling decrees that minerals found in bulk in their natural state should be shipped in bulk and the general freighter is designed to comply with this axiom. Specialist vessels are designed for transporting one specific commodity such as cement, oil, pulpwood, dairy products, package freight, etc., but the all rounder carrying grain, ore, coal, limestone, etc., in bulk displays the essential features of versatility consistent with economical design. She can obtain access to a great number of navigationally difficult ports and be loaded and unloaded with diverse pieces of equipment.

Ore and grain constitute the major cargoes for the Canadian owned vessel. Due to source distribution of the cargoes available large lakers usually proceed upbound from Lake Erie to Lake Superior in ballast and downbound in the loaded condition. From Lake Erie to the River St. Lawrence they can be in the loaded condition each way although some ships are tied to either the ore or grain trades in which case they may be loaded only on the up or downbound trips respectively.

A round trip involving these cargoes might commence at Seven Islands loaded with ore which is carried up river to an American port on Lake Erie, say Cleveland, for discharge and thence in ballast to Port Arthur on Lake Superior to load grain for Quebec from which she returns to Seven Islands in ballast. There are many variations but this is a practical trip covering almost the full extent of the service area and gives the general conditions of loading and ballasting investigated in Part III. The average time taken by a 16 mile per hour vessel for the 3,550 miles voyage is around 17 days, during which time about 24,000 tons of ore are moved 1,046 miles and 1,180,000 cubic ft. of grain 1,370 miles. The overall time is, of course, subject to variation through delay at loading and discharging docks, in the Seaway, weather conditions and to tides in the lower river.

Operational Restrictions

The harbour at Seven Islands was built as an ocean terminal and presents no problem in dimensions or draught but a large laker contemplating a loaded voyage to Lake Erie is limited in draught to 25 ft.* (FW) by the St. Lawrence Section of the Seaway and although the Welland Ship Canal will also be encountered the allowable draught in this section is 25 ft. 6 in. She would load to about 24 ft. 6 in. (SW) with slight trim aft so that when St. Lambert Lock is reached with

* Increased to 25 ft. 6 in., April 1963.

Plate 3

Frankcliffe Hall in the Welland Canal

partial fuel and stores consumed she will trim to a level keel at the mandatory 25 ft.* draught. The operators have become quite expert at obtaining this limiting draught taking into account an inch or so of sag and without utilising builders' loading data.

On hot sunny days in summer the deck can become hot enough to produce hog which increases forward and after draughts and it is then not uncommon for the crew to play hoses on the deck plating before passing through the locks. Vessels occasionally have to lie alongside the canal approach. wall to await the cool of evening before permission to proceed is granted. The 25 ft.* maximum must not be exceeded forward or aft which of course tends to encourage the use of some sag and the restraining procedures controlling sag will be discussed in Part III.

The American ports on Lake Erie receiving ore cargoes were developed to take Lake Superior ore from downbound vessels restricted in recent years to about 23 ft. 6 in., and previously to 20 ft., by the channel at Sault Ste. Marie. Harbour improvements since the Seaway opened have mitigated the draught problem for the upbound loaded vessel.

Lake-head grain elevators, however, can barely accommodate 23 ft. 6 in. draught and are much less than this close to the face of the elevators. Consequently vessels are forced to keep working out from the wall as loading proceeds to ensure that they are not sitting hard on the bottom. The outreach of elevator equipment in turn restricts the distance the vessel can be worked off the wall and there are few places where these large ships are able to load to 25 ft. draught but dredging projects are in hand.

Ore loading equipment at upper lake ports restricts beam, depth and draught. At the older docks ore is normally stored in elevated bins at the loading berths and is fed into the ships by gravity through inclined chutes. The height and angle of the chutes and their ability to throw ore at least to the centre of the ship should fix the maximum depth and beam but in practice this is not the case. Loading facilities vary considerably the only common feature being the spacing of the chutes at multiples of 12 ft. Ships of 65 ft. beam and about 32 ft. deep were at one time considered optimum but some ten years ago beams were increased to 70 ft. and depths to 37 ft. Now ships of the Seaway size of 75 ft. beam and 39 ft. deep are pushing their way under the ore spouts. At some terminals because of the restricted throw of the spouts ships require to be reversed after being partly loaded so that loading can be completed. The vertical distance from the dock bottom to the spouts is sometimes little more than the depth of the ship, forcing the vessel to hold a near constant draught by berthing in ballast and maintaining displacement by ejecting ballast in time with loading. All the 39 ft. deep ships operate in variations of this procedure and some loading ports just cannot handle them.

Generally, draught at upper lake loading ports is restricted to the "pre-Seaway" depth of 23 ft. 6 in. of the Sault Ste. Marie channel and some have from 12 ft. to 20 ft. of water. Deepening by dredging and expansion and modernisation of equipment is proceeding which is helping to ease the situation for the large ship but facilities are inadequate to realise fully the potentialities of the present dimensions. At many ports dredging could easily undermine elevator piling and practical draughts cannot be attained close to the quay walls without extensive replacement of sub-structures.

Unloading facilities have almost entirely disappeared from coal and stone docks and these commodities are becoming the monopoly of the self-unloader.

Length of ship is seldom restricted at loading and unloading ports except that long ships may occasionally experience some difficulty in manœuvring.

* The maximum draught for transit through the St. Lawrence Section of the Seaway was increased in April 1963 by six inches to 25 ft. 6 in. and this permits maximum sized upbound lakers with cargoes for ports on Lakes Ontario and Erie to utilise this new draught in the midsummer and summer seasons. In the intermediate and winter seasons, however, loading will be limited to the applicable load line marks amidships. The first design for a large laker with scantlings suitable for the 25 ft. 6 in. draught in the intermediate season has already been dealt with in the Montreal Office.

Principal Dimensions

There is a mass of conflicting operational data, with associated theories and assumptions, shrouding laker design and yet with clear and logical thinking it becomes all too evident that the cardinal characteristics, length, breadth, depth and draught are only affected by two incompatible factors which are the owner's desire to have the largest possible ships (Ref. 5) and the St. Lawrence Seaway Authority's limit on dimensions for transit through the Seaway.

No loading port authority has yet issued a mandatory instruction regarding the limit of size the port is willing to handle and so ship dimensions have been increased out of step with port renovations, extensions and dredging. It would appear that length and breadth are now established for the foreseeable future and there may be a reasonable chance some port authorities will catch up.

Depth is not directly restricted by the Seaway Authority but its relationship to freeboard and draught determines the minimum value while the height of ore spouts at lake-head ports sets a non-applicatory maximum which, in theory, is less than the minimum but in practice must admit to being the same. So, too deep ships scrape the bottom while struggling to be loaded under ore spouts built for a bygone age.

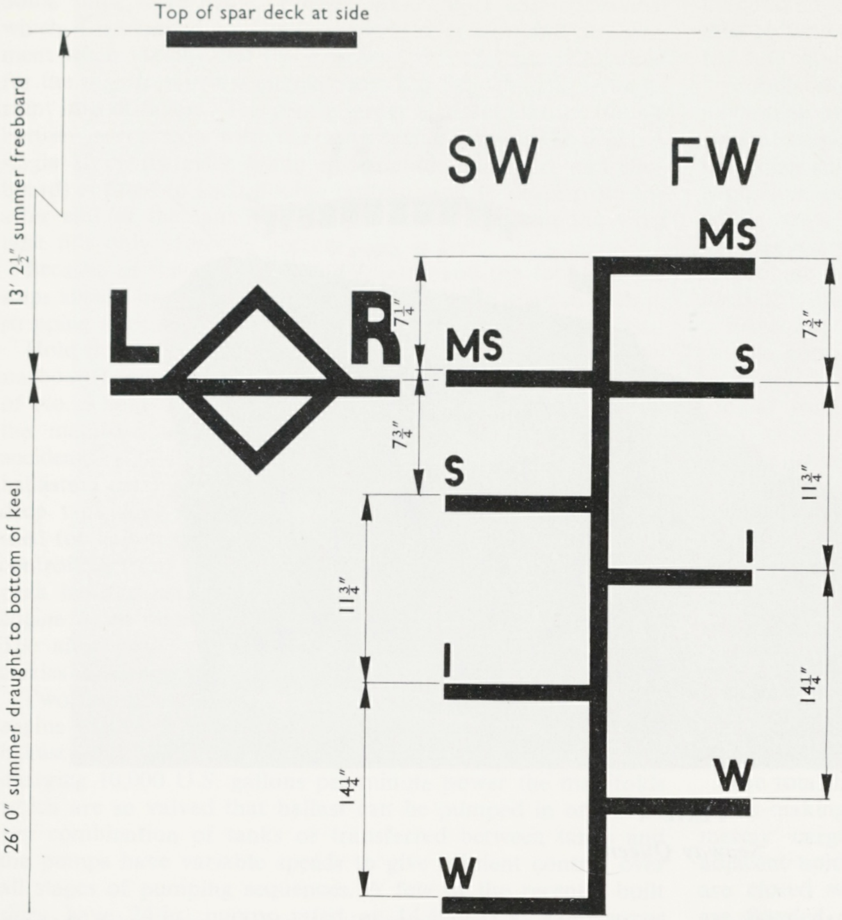
Draught is closely related to deadweight which in turn is married to profit and therefore receives the consideration and respect such high position deserves. The 25 ft.* maximum draught of the St. Lawrence Section of the Seaway is set as the starting off spot in structural design and must be minimally obtained in the intermediate season. This ensures that for over 80 per cent of the operational season the maximum permissible seaway draught can be utilised.

In order to make the maximum draught apply for the full operational period, that is to include the unpredictably limited winter season, the scantlings draught would require to be increased by 1 ft. $3\frac{1}{2}$ in. and the depth by about 2 ft. The inconvenience in loading with this depth and the additional weight necessary to provide the longitudinal strength required for the draught do not justify adopting the increase.

The minimum summer moulded draught on which the scantlings and structural arrangements are based is therefore 25 ft. $10\frac{3}{4}$ in. When the $11\frac{1}{4}$ in. intermediate deduction to draught and the keel thickness are taken into account the 25 ft. extreme draught is obtained. Having in mind that the Seaway draught may increase to 25 ft. 6 in. in the near but as yet unstated future most builders aim at a draught slightly in excess of this minimum. In order to obtain the minimum geometrical freeboard corresponding to a summer moulded draught of 26 ft. the depth of the vessel requires to be at least 39 ft.

Therefore if it is intended to operate east of the Seaway which includes Montreal harbour, a vessel whose moulded dimensions are 710 ft. × 75 ft. × 39 ft. designed for a fresh water load draught of 26 ft. in the summer season appears

* Increased to 25 ft. 6 in., April 1963.



FREEBOARD MARKINGS

for a ship making voyages on Lakes or Rivers

MS—Midsummer:	Applicable 16th May to 15th September.
S--Summer:	Applicable 1st to 15th May and 16th to 30th September when midsummer load line assigned, but for other ships is applicable 1st May to 30th September.
I—Intermediate:	Applicable 16th to 30th April and 1st to 31st October.
W—Winter:	Applicable 1st November to 15th April following year.
FW—Fresh Water:	Denotes seasonal maximum draughts to which ships may be loaded in Great Lakes and fresh water area of River St. Lawrence.
SW—Salt Water:	Denotes seasonal maximum draughts for operation in salt water in lower River St. Lawrence.

NOTE: The moulded draught in summer (FW) is that used to determine scantlings.

FIG. 1
Load line markings for 710 ft. × 75 ft. × 39 ft. laker

to be the ideal within the limiting conditions imposed. Most of the recently-built Canadian large lakers approach these dimensions.

It will be noted both depth and beam are small when related to length and that the proportion of length to depth is 18·2. This ratio was established over 50 years ago and can hardly now be considered as improper practice. The previously mentioned *Lemoyne* which is sailing successfully to-day is 21·4 depths to length and is the highest ratio from 1926.

LOAD LINES

The Canada Shipping Act, 1934, made compulsory, with some exceptions, the assignment of load lines to ships employed on lakes or rivers and subsequently this law was made effective " as of the first day of October, 1937". Up to that time there was no statutory requirements for draught but this in itself was not so undesirable in the larger ships as might be expected since draughts on the whole were determined by the restricted waterways in which the vessels operated. Much more to the point, however, was the absence of mandatory standards of closing appliances and of longitudinal and transverse strength. This explains, in some degree, the present wide divergence existing between lake and sea-going practices in many aspects of shipping on the Great Lakes.

Unlike sea-going ships whose freeboards are universally related to salt water, laker load lines are governed by the Canadian Rules for ships making voyages on lakes or rivers which are based on operation in fresh water and for recognition purposes, a "diamond" is marked on the ships' sides to distinguish lake load lines from the international "disc". The complete load line markings which could be assigned to a maximum sized laker are illustrated in Fig. 1 together with dates and explanatory notes regarding the application of the marks. The salt water lines, made necessary by the Seaway, are a recent addition to the "grid" of the larger ships to facilitate operation down river and are additions to the freeboards which is the reverse of the sea-going fresh water mark.

The Lakes or Rivers Rules for computing corrections to the basic summer freeboards are identical to those of the international regulations with two exceptions. The first is in the tabulated minimum summer freeboards and the differences in the tables are essentially to provide the lake ship with the fresh water draught she would obtain if her summer freeboard had been computed in accordance with the international rules with the fresh water allowance added to the corresponding draught. The second difference favours the large slender ships and is in the correction for depth under Rule 42(3) permitting ships over 350 ft. in length with no midship superstructure to be less than the standard depth of L over 15

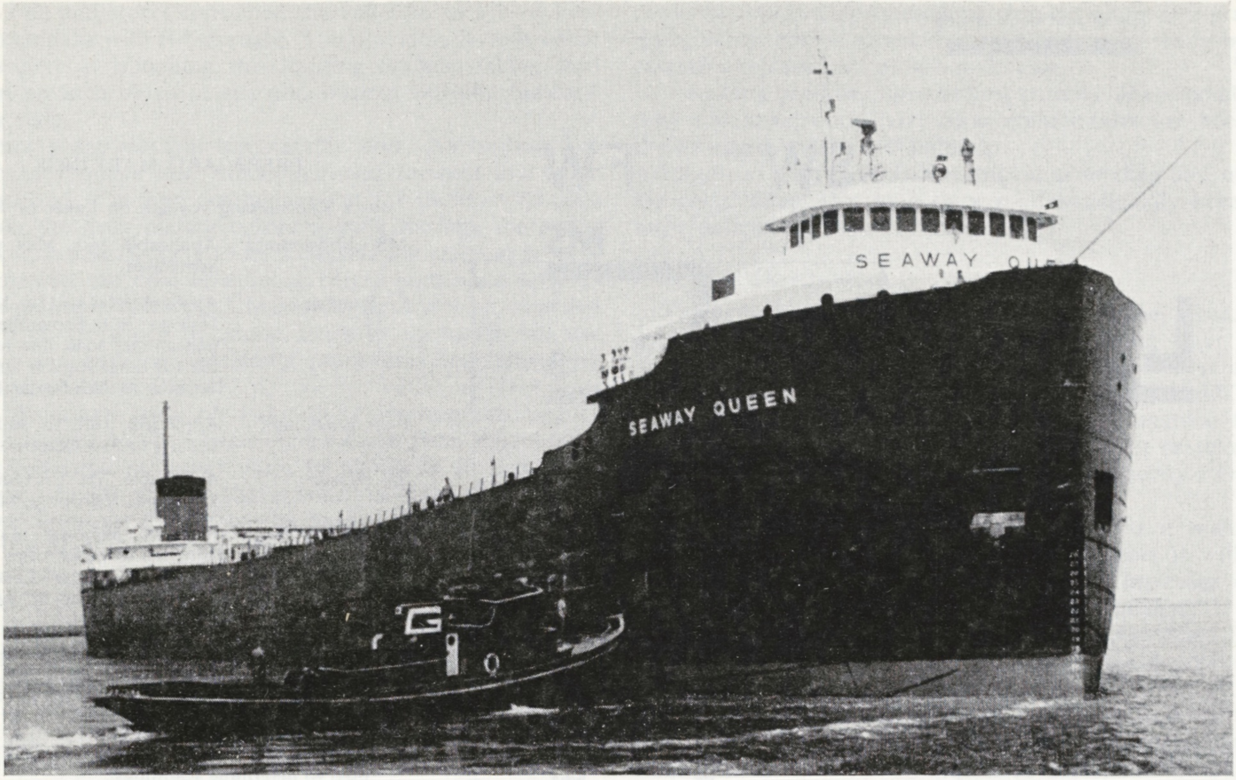


PLATE 4

Seaway Queen

without freeboard penalty. The 710-ft. laker of 18 depths to length gains about 24 in. in draught by this rule.

In ships where the length to depth ratio is less than 15, the summer draughts in fresh water computed in accordance with the Lakes or Rivers Rules and the International Load Line Convention Rules are generally the same and so also are the two summer draughts in salt water. This is consistent with Rule 99 of the Lakes or Rivers Regulations which permit any sea-going ship with full international load lines to load to TF, F, S and W markings when in the fresh water lakes during the midsummer, summer, intermediate and winter seasons respectively. The effect of this rule is to allow sea-going ships to operate, in most cases, at the draughts which they would be granted if their freeboards were computed in accordance with the Lakes or Rivers Rules.

Ballasting Arrangements

Large lakers usually carry a full load of one type of cargo on each loaded trip: that is to say, they carry ore or grain or coal or stone but never a mixture of two or more. This simplifies loading and eliminates the adverse conditions which often arise with mixed cargoes. So far as longitudinal strength is concerned loading can always be controlled within the limits imposed by draught considerations. Thus the loaded conditions of these ships when in the open lakes are rarely, if ever, such that improper distribution might cause concern.

When in the light condition, however, a different set of conditions arise and it becomes apparent that the most important single feature in the operation of large lakers is in the efficiency and versatility of their ballasting facilities and in the unfavourable conditions which could arise through improper use. An outline description of the ballasting arrangements of a 710 ft. laker is therefore deemed necessary for the better understanding of the investigation carried out in Part III.

Efficient and speedy filling and discharge of ballast is of prime importance. Loading of cargo can usually be done in a few hours and since heights and draughts under loading spouts are so restricted at the head of the lakes ballast must be retained until the ship is safely docked and loading imminent. Complete discharge of ballast must then be accomplished in less time than is required to load. The permanent ballast tanks usually take from 11,000 tons to 12,000 tons of fresh water, that is 42 per cent to 46 per cent of the deadweight, and must be capable of discharge in two and a half to three hours for optimum control when loading.

The total weight of ballast taken aboard when under way is dictated by weather conditions and can vary between 5,500 tons and 18,000 tons when hold flooding is included. Conditions Nos. 1 and 47, Table 10, are based on approximate minimum and maximum displacements in ballast.

When unloading, ballasting is always commenced so as to be complete when all cargo is discharged to ensure speedy turn round. Lake operators are indefatigable in avoiding time loss.

A typical midship section of a large laker is shown in Fig. 5 and an outline general arrangement in Fig. 6. There are six double bottom and wing tanks (port and starboard) extending for the length of the hold space and the tank end bulkheads coincide with the screen bulkheads in the holds.

Some ships have eight of these tanks (port and starboard) which it is claimed give better control over trim and displacement when loading but the increased size and space required for the manifold in the engine room has brought this arrangement into disfavour. The centre girder is intact and the double bottom is common with the wing tank to the main deck. A single 12-in. diameter filling and suction line (port and starboard) is fitted to each double bottom tank to drain from the after end of the tank near to the centre line and the wing tank fills only after the bottom tank is full.

Because of the efficiency of discharge and the fact that the ships always have substantial trim by the stern when in ballast, stripping lines are rarely fitted.

Hold flooding is done through a 12- or 14-in. line (port and starboard) led to a drain well in the after outboard corner of No. 6 hold and when carrying cargo the line is blanked at the manifold to eliminate the disastrous consequences of accidentally flooding a hold full of grain. To complete the ballasting arrangements a single 6-in. line serves the forward deep tank and fore peak. The fore peak which is seldom used for ballast can be isolated from this line by a gate valve controlled from the forecastle deck. Occasionally the fore peak has the luxury of a ballast line to itself but since the engine room manifold is more than 600 ft. away this is rare. The after peak is also seldom used for ballast and usually carries fresh or sanitary water but sometimes is a dry space.

Two enormous manifolds are fitted at the fore end of the engine room, one port and one starboard, to control the ballast lines. Two 20-in. vertical pumps each capable of discharging 10,000 U.S. gallons per minute power the manifolds which are so valved that ballast can be pumped in or out of any combination of tanks or transferred between tanks and the pumps have variable speeds to give efficient control over all stages of pumping sequences. A few of the recently built ships have 24-in. pumps rated at 14,400 U.S. gallons per minute which means, on paper at least, that a total capacity of 12,000 tons of ballast can be pumped overboard below the light load line in 1 hour 50 minutes. It is difficult to imagine that this rate can be matched by any vessel afloat except possibly by a floating dock. Considerable damage by erosion has been caused to canal banks and dock piling by the discharge from these high rated pumps and their use has been limited by some authorities when the vessels are in confined waters.

Control of ballasting is carried to exceedingly fine limits. At the control console in the engine room and in the spar deck passage way in the forecastle indicators are fitted which, by visual inspection, give at any instant the capacities of each of the 14 ballast tanks and also the vessel's forward, midship and aft draughts. In addition the current state of about 12 tanks comprising bunkers, diesel oil, fresh and distilled water, etc., are indicated at the control platform.

The ballast manifolds occupy 30 ft. by 4 ft. of the tank top space in the engine room and the power operated valves are centrally controlled by press-buttons with manual operation used only in emergency.

Hold Flooding

Since ore and grain are the principal cargoes structural design is affected by each. The ratio of hold space to ballast is compromised to accommodate these commodities to best advantage and obviously the ship cannot be perfect for both. Ore occupies less than half of the available hold space at load draught while grain sometimes fills the hold space before the maximum permissible draught is attained. A third factor in this equation is the necessity for sufficient ballast capacity for safe navigation in the light condition.

Vessels engaged primarily in the ore trade have a distinct advantage and can give up hold space to ballast while the general cargo and grain carrier is more critical in its sacrifice. Canadian ships engaged in a definite ore trade, cannot ignore a possible grain cargo, which is much better than ballast, on return trips and therefore retain their semblance of grain carriers but with one leg of the voyage assured they do tend to provide more ballast tank capacity. They fall short of the ideal for ore but the compromise is economically sound.

Freighters intended for ore traffic might have 6 ft. deep double bottoms while 5 ft. and even 4 ft. 6 in. depths might be adopted where grain is the main cargo. Similarly the longitudinal wing tank bulkheads are sloped inboard from the main deck to the tank top in ore carriers and are nearer to vertical in the general purpose vessel.

In attempting to design for full draughts when carrying grain the tank capacities available for water ballast are so reduced as to be insufficient and from this arises the need for hold flooding. The origin of hold flooding is lost in the past but the practice was universal in the 255 ft. canaller and is accepted on the lakes as an indispensable condition.

Ballasting is, of course, the responsibility of the Master and there is little evidence that the methods adopted are unduly detrimental to the ship's structure. Custom and usage have created a distinct pattern for ballasting although individual skippers have their own closely allied variations.

The total hold space is usually divided by five screen bulkheads making six separate compartments and the screens are merely cargo divisions having drainage tunnels, linking adjacent holds, at the double bottom tank top. These tunnels are closed when carrying cargo and open when the holds are flooded allowing water ballast which is filled from the single inlet (port and starboard) in No. 6 hold, to find its own level throughout the entire length of the cargo space. In the past the forward engine room bulkhead was sometimes marked on the hold side with a scale in feet above the tank top so that depth of ballast was readily discernible but this practice has been discontinued in the modern large ships. With good propeller immersion in ballast the vessel is trimmed by the stern and with about 9 ft. of water on the after hold bulkhead the depth of water will taper forward to nil somewhere in No. 1 hold. Condition No. 42, Table 10, represents this condition when Nos. 1 to 6 ballast tanks are full.

Corrosion

Corrosion of the hulls of lake ships is not a serious problem and wear and tear repairs are usually limited to inner bottom and side tanks due to loading and discharge. Inevitably, grounding and quay wall contact make up a large part of damages. Machinery in the lake ship generally becomes obsolete or wears out before the hull whereas in sea-going ships especially tankers the reverse is generally true.

This absence of corrosion makes for a long useful life which is favourable to high initial cost and there is possibly no type of commercially operated ship in which operational economics is permitted to play such a major part. Many intangible factors usually prevent sea-going ships from being the optimum economical design but this is definitely not the case with the large laker. Only functional characteristics related to maximum overall efficiency of operation are permitted to influence design.

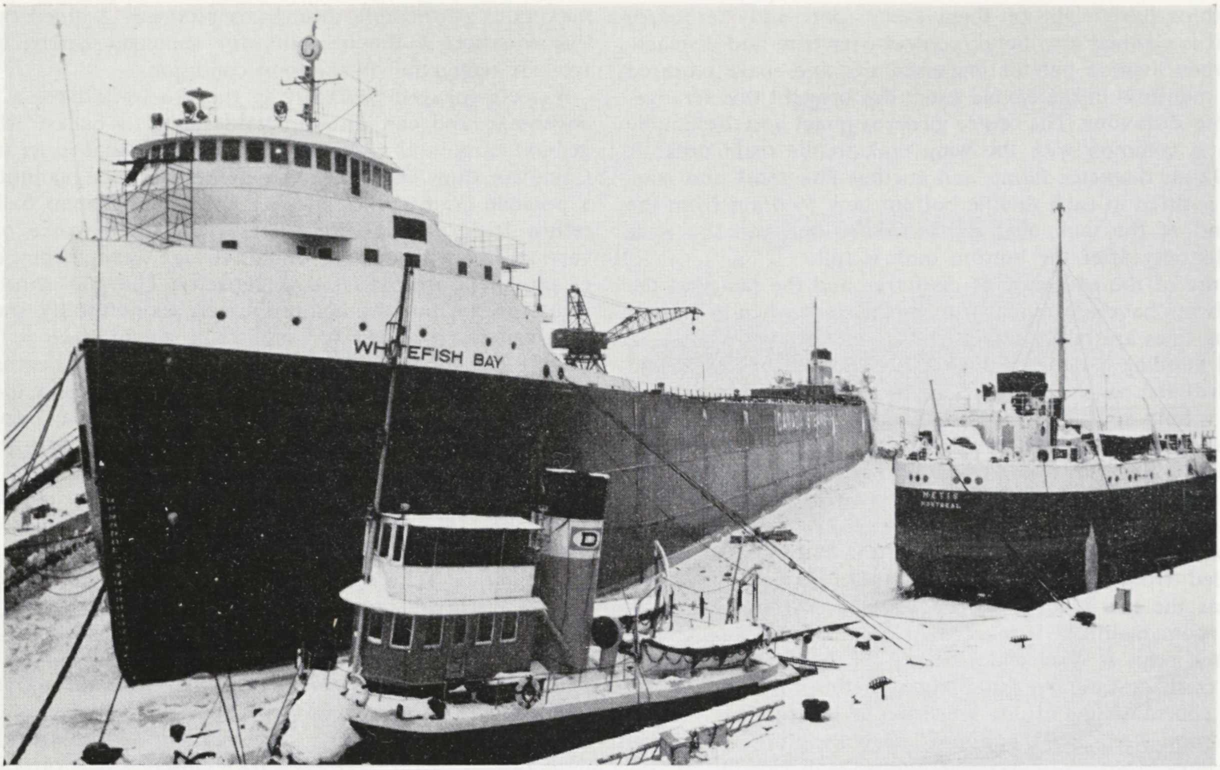


PLATE 5
Whitefish Bay fitting out in winter

By aesthetic standards modern large lakers may lack beauty but in their undisputed mastery of difficult conditions they have acquired a dignity of deportment and air of majesty which can be felt by those familiar with the vital role these vessels play in the transport blood stream of Canada.

Five photographs of Canadian-owned and built lakers, each over 700 ft. in length, are included to represent these denizens of North America's fresh water lakes and rivers.

PART II—MOMENTS AND STRESSES

Longitudinal Bending Moments

The Society's method of assessing longitudinal bending moments was introduced in 1947 by J. M. Murray (Ref. 7) and papers on the subject were subsequently presented to various technical institutions (Refs. 8 and 9) and to the Staff Association (Ref. 10). This method was developed for use with sea-going ships, principally oil tankers and dry cargo carriers, giving excellent results in these cases and for the purpose of the present paper it was decided to examine the extent to which this system might be applicable to long slender lake ships so that practical analysis of the infinite variety of ballasted and loaded conditions of these ships might be possible. In order to develop this theme in the preliminary stages some repetition of the salient features of Mr. Murray's method is necessary.

The total longitudinal bending moment induced in a ship in service is made up of two components; one is the still water bending moment due to the disposition of weights acting in opposition to the still water buoyancy and the other is the moment due to waves. The magnitude of the still water bending moment can be assessed with reasonable accuracy from the design of ship and distribution of loading, whilst the theoretically computed wave bending moment tends to be academic and lacking in intrinsic reality.

Both of these components will be examined more closely to show that consideration of longitudinal stresses in large lakers can be based solely on still water bending moments thus eliminating the controversial, and apparently indeterminate, subject of the relativeness of Great Lakes storm waves to those of the oceans of the world.

Still Water Bending Moments

The maximum bending moment in still water is usually near to amidships and, the ship being in equilibrium, the differences of moments of weight and of buoyancy about midships will give the still water bending moment at midships.

Moments of weight for the forward and after bodies are calculated from the particulars of the ship to be investigated including the disposition of her cargo or ballast and so the mean moment of weight about midships obtained. This principle is obviously applicable to any ship and can be used in the large laker.

The calculation to determine the mean position of the L.C.G. of the forward and after bodies of the hull weight is long and tedious and for sea-going ships use is sometimes made of the approximate method of dividing the length of ship into three equal parts and raising ordinates, from forward, of ·566, 1·195 and ·653. This weight curve was originally devised for fine merchant ships (Ref. 11) and is

hardly representative of the large laker. Bennett in considering the laker problem in 1929 (Ref. 12) departed from this sea-going practice and, being influenced by the long parallel middle body, adopted ordinates of ·572 at F.P., 1·125 at $\frac{1}{4}$ length forward and aft and ·676 at A.P. This weight curve is a closer approximation to the actual weight distribution and gives an L.C.G. of total weight of ·0054L aft of midships or 3·84 ft. in a length of 710 ft.

From investigation of particulars supplied by builders of modern large lakers it was found that the total hull weight from keel to spar deck is practically balanced about amidships with the L.C.G. being less than ·002L forward of midships. In determining the mean L.C.G. of the forward and after bodies therefore a symmetrical weight distribution about amidships was adopted having ordinates of ·7 at each end and 1·1 at the $\frac{1}{4}$ lengths forward and aft (Fig. 2). When average moments of the steel weights of poop, forecastle, funnel, outfit, etc., were included with this weight curve it was found that

mean L.C.G. of forward and after bodies = ·245L.

This expression gave reasonable approximation to builders' calculated moments of light ship excluding machinery.

Moments of buoyancy for sea-going ships may be obtained using Murray's formula (Ref. 7) which gives the mean moment of buoyancy of the forward and after bodies thus:—

mean buoyancy moment = (D/2) × mean L.C.B. of forward and after bodies

where D = displacement at mean draught amidships.

The mean L.C.B. of the forward and after bodies is expressed in the form aL where "a" is a function of block coefficient and L is the B.P. length of ship. The values of "a" were derived from analyses of a large number of sea-going ships of normal proportions having varying block coefficients and the values obtained give good results for draughts between ·03L and ·06L with trims not exceeding ·01L.

When considering large lakers the following four points arise making necessary a reassessment of the values of "a" for these ships.

1. Operational draughts rarely exceed ·035L and may be as low as ·015L.
2. Operational trims can be up to about ·03L.
3. Block coefficients are invariably within ± 2 per cent of ·865 at load draught.
4. Proportions are abnormal when related to sea-going practice.

Therefore with suitably modified values of "a" the mean buoyancy moment of the forward and after bodies for the large laker can be determined by the Murray method.

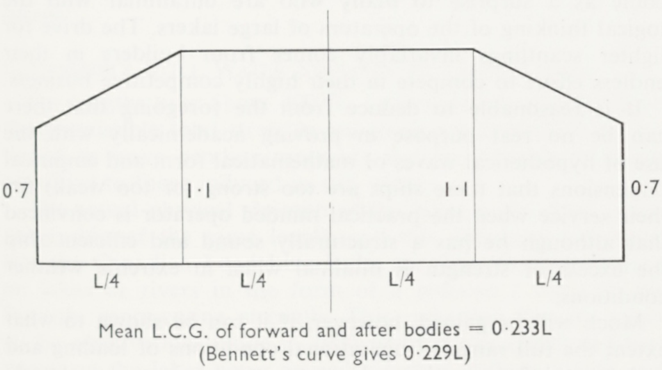


FIG. 2

Hull weight distribution, keel to spar deck

Assessment of still water bending moments can now be summarised as follows:—

where MW = mean moment of weight
MB = mean moment of buoyancy
Wf = moment of weight forward of midships
Wa = moment of weight aft of midships

then MW = (Wf + Wa) /2
and MB = (D/2) × a L
where "a" is a coefficient derived from Table I.

Finally SWBM = MW − MB (hogging)
= MB − MW (sagging)

VALUES OF "A"

Using bonjean curves supplied by Canadian builders for recently built lakers between 700 ft. and 710 ft. B.P. having length to depth ratios of about 18, moments of buoyancy for the forward and after bodies were computed for draughts and trims covering the operational range. From these particulars values of "a" were obtained and tabulated in Table I.

It was found that within the operational range, trim had considerable effect at small draughts with reduced effect as draught increased. Small variations in block coefficient produced negligible changes in "a" which for practical purposes need only be related to draught and trim.

The position of the L.C.B. of the whole displacement is invariably well forward of amidships for all draughts (*see* Fig. 7) but the actual distances vary in individual ships. This feature was found to affect the value of "a" and average values are tabulated.

TABLE 1

Values of "a" for mean LCB of fore and aft bodies

Mean Draught	Level Keel	Trim by Stern (deduct)	Trim by Head (add)
·04L	·2220	Nil	Nil
·03L	·2190	Nil	Nil
·02L	·2140	·1 t/L	·2 t/L
·015L	·2110	·2 t/L	·4 t/L
·01L	·2070	·3 t/L	·7 t/L
·005L	·2000	·4 t/L	·14 t/L

L = B.P. length in feet and t = trim on length L in feet.

It should be noted that trim aft reduces, while trim by the head increases, the mean draught value of "a". Provided the table is used for vessels of standard lake fulness having length to depth ratios of around 18 practical accuracy will be obtained for any operational trim down to a mean draught of about ·01L.

The mean position of the L.C.B. of the fore and after bodies can, of course, be determined by calculation if desired but use of the foregoing table eliminates much of the tedium.

WAVE BENDING MOMENTS

The first systematical investigation into operational weather conditions of lake vessels was carried out in 1920 (Ref. 16). Masters were instructed to procure data on the length, height and period of the longest waves observable and were provided

with prepared forms and diagrams for the purpose. During this operation 187 observations, of which 115 were of wave lengths and 72 of wave heights, were made in the fall of 1920 between 20th September and 23rd November and Prof. Sadler, Chairman of the American Committee appointed to investigate and analyse these data, recorded the statement, "although all possible care was taken on the part of the observers, there was still a chance of error" (Ref. 13).

Except for some extant data made by six lake-side stations of the United States Weather Bureau, little or no information was available on this subject before 1920 and in addition no methodical enquiry had been made into the strength of vessels on the lakes.

By present-day standards of measurement these observations, extremely well intentioned as they were, leave a great deal to be desired. Sadler and Lindblad considered on the whole, however, that the data so obtained were sufficiently reliable to permit them to carry out a theoretical investigation into the strength of these ships and to propose a standard for vessels up to 600 ft. in length. That this standard, based on a nine-week period of operation, was successful is now an historical fact indicating exceptional foresight and excellent judgment must have been contributed in its formulation.

Very little additional evidence on weather conditions was recorded between 1920 and 1934 when a slightly modified and reduced standard of strength for lake ships became part of the Canada Shipping Act. Now after nearly 30 years of experience in the operation of ships complying with this standard there is no record of a vessel having been lost through main structural weakness implying at least, that lakers are strong enough for their service. Some ships suffer fracturing of their spar deck at hatch corners mainly in the midship region and one is known to have fractured across the stringer plate which in turn implies that the margin of strength in hand is small.

So far as the Author can ascertain failure of the bottom plating due to structural weakness has never been recorded.

There is the case of the 640-ft. *Carl D. Bradley* which was alleged to have been lost through structural deficiency but the evidence available is inconclusive. There appears no doubt, however, that the spar deck fractured near to amidships and opened up. The ship had given successful service for 30 years and was making her 46th trip of the year having covered 24,000 miles during the 1958 navigational season when the disaster occurred on 18th November.

The point of greatest interest so far as this paper is concerned, however, is the state of the weather which had been building up for a few days previously. The wind was from the south-west blowing the 300 miles length of Lake Michigan and, at the time of the casualty, had reached a velocity of greater than 60 miles per hour producing waves in the north end of the lake of 20 ft. in height with the occasional wave probably much higher.

The "Bradley" was a self unloader with propulsive machinery aft and unloading equipment forward. She was alleged to have been carrying about 9,000 tons of water ballast amounting to half of her deadweight capacity and her earlier service record indicates that the longitudinal stresses produced by this condition must have been moderate. Sufficient data are not available to the Author to investigate the longitudinal stresses but with the heavy equipment forward, the tensile stress at the deck due to hogging would be higher than that in a similar bulk freighter in ballast without unloading gear and because of internal conveyor belt equipment, hold flooding, which eases hog, was not practicable. It can be concluded, therefore, that the magnitude of the longitudinal bending moments near to amidships although producing no previous detrimental results, must have been greater than the average for ships of this length. This fact would have a contributory effect in association with the circumstances immediately preceding the loss.

The vessel was in transit from Buffington, Indiana, at the south end of Lake Michigan to her home port at Rogers City on Lake Huron and during her voyage northwards she was running before the wind. About half an hour before the disaster she was reported to have been behaving well, which would imply her long hull was heaving and surging and generally producing the visible movements of successful combat with the elements which steamboat men take for granted as part of the nature of things. Towards the north end when off Boulder Reef and Beaver Island she must commence the long arc which will bring her on an easterly course for the Straits of Mackinac and it is significant that soon after changing course, carrying her obliquely across this wild sea, she was to become a total loss.

Unlike navigation in open seas, large lake ships have often no alternative in the course they can take. In heavy weather they will attempt to complete their voyage and in a restricted expanse of water this often necessitates taking a direction across a sea which the ocean-going ship would abhor. During this manœuvre the twisting moment inflicted on the long slender ships must be quite considerable when the weather is severe and it is considered that the *Carl D. Bradley* was a case in point in which a combination of adverse conditions summed up to more than the ship could stand.

Responsible representatives of lake shipowners are of the opinion that this manœuvre, where violent twisting of the hull is induced, creates the greatest strains the ships ever encounter in service and is the condition which normally produces fracturing at hatchway corners when this occurs. While they may consider their ships, whether loaded or in ballast, amply strong to ride out any sea experienced on the lakes, concensus of opinion of these experts would appear to be that the present margin must be maintained if spar deck damage, at present relatively small, is not to increase.

In other words, owners, on the advice of their naval architects are not willing to gamble the gain accruing from saving a few square inches of strength deck sectional area against a probable increase in lost time for deck repairs during the short operational season.

This admirable viewpoint from successful enterprise may come as a surprise to many who are unfamiliar with the logical thinking of the operators of large lakers. The drive for lighter scantlings invariably comes from builders in their endless effort to compete in their highly competitive business.

It is reasonable to deduce from the foregoing that there can be no real purpose in proving academically with the use of hypothetical waves of mathematical form and empirical dimensions that these ships are too strong (or too weak) for their service when the practical minded operator is convinced that although he has a structurally sound and efficient ship the excess of strength is minimal when in extreme weather conditions.

Much will be gained, however, if it can be shown to what extent the full range of operational conditions of loading and ballasting influence the still water stress since, as already mentioned, this is the component of total stress which can be assessed with some degree of accuracy.

The magnitude of calculated wave bending moments in sea-going ships is dependent on hull form which may vary considerably from ship to ship and each case must be determined individually. For practical purposes the laker is of standard hull form and it can be said that for lakers of equal length the theoretical maximum wave bending moments must also be equal. There is a range of operational draughts at any given length of ship but Murray was satisfied from investigation that when the Smith correction is taken into account there is no appreciable change in wave bending moment for draught.

If then the wave bending moment is constant for any draught at a particular length of ship, the maximum permissible still water bending moment will also be constant, since the two together make up the total permissible moment.

A sound basis of comparison of longitudinal strength of hull structures of large lakers is therefore available from consideration of the static disposition of weights relative to buoyancy in still water and the crux of the matter lies in determining the proportion of total permissible longitudinal stress which can be safely allocated to still water bending moment while leaving sufficient balance to successfully assimilate the additional moment caused by lake storm waves.

Permissible Total Longitudinal Stress

The large bulk freighter on the lakes is designed to carry ore and since this is her most onerous cargo her longitudinal strength was compared with that of a sea-going ore carrier. Their behaviour in service and general characteristics bear close resemblance so far as bending moments are concerned. They both sag if uniformly loaded and hog in ballast and the disposition of their cargo, ballast and machinery space has a similarity. They are each longitudinally framed at deck and bottom and have two longitudinal bulkheads.

The longitudinal strength of sea-going ore carriers is based on the requirements for tankers of similar dimensions and the permissible total longitudinal stresses are published in Refs. 8 and 14. These are compared in Fig. 4 with the stress of $\sqrt[3]{L}$ used by Dr. Sadler in the 1920 investigation of lakers up to 600 ft. in length.

The Society's total stresses are based on the examination of the strength of ships giving successful operational service as well as those which gave trouble and are related to general experience rather than optimum design (Ref. 8). Since they are related to this experience and are not too far removed from those employed by Sadler it seems reasonable to assume that such stresses can be successfully accommodated by large lakers.

Permissible Still Water Stresses in Large Lakers

For lengths over about 400 ft., approximately two-thirds of the permissible total stress in sea-going ore carriers is apportioned to wave bending moments and the balance of about one-third to those induced in still water.

The actual physical strength of the laker is less than a sea-going ship of the same length and this difference is written into the Canadian Load Line Rules for ships making voyages on lakes or rivers in the form of a reduced f value in the formula representing strength. This reduction has its origin in the observations made in 1920 when it was concluded that waves on the lakes never exceed a certain definite length and thus as ship length increases the divergence of strength also increases. Expressed differently, it can be said that the laker requires less of its total strength to combat wave stress than does the sea-going ship and complementary to this, the permissible still water stress may be higher.

If it is conceded that a large laker, built of the same material including Grade D steel where applicable, can withstand the same total longitudinal stress at the deck and bottom as a sea-going ore carrier then she can successfully operate with the sea-going still water stress increased in proportion to the reduction in wave stress experienced in service.

The Committee set up by Great Lakes shipping interests to examine their local load line problem used the sea-going load line standard of strength as their yardstick. That the two types of ship should have this common origin is not altogether a fortunate state of affairs as many of the original limitations attached to the adoption of each of these standards are now lost in time and some ignored. It is not intended, nor is it necessary, to enquire into these limitations except to emphasise that the establishment of any basis of comparison of the two standards is fraught with compromise and can never be absolute.

Both these standards with marginal variations, stood the test of time in their own spheres and it is only in the last few years that modern advancement in structural design and arrangements has produced ships for which the ocean standard appears too limited in scope to be acceptable as the universal sea-going minimum.

The present-day classification requirement for longitudinal strength of lakers is related to fBd and so varies directly with draught. The evolution of determining the minimum strength required for sea-going ore carriers and tankers is proceeding independently of draught although a foothold in draught is still retained. This modern approach to the latter type of vessel makes comparison even more unreliable and possibly even undesirable.

Nevertheless, in view of custom, tradition and common origin the only feasible mode of comparison is to endeavour to relate the standards by their f values and this was done by accepting the sea-going f value as unity and expressing an "equivalent" lake f value as a ratio of this unity. The Lake Load Line Rules do not provide f values for vessels in excess of 620 ft. in length and similarly there is no statutory standard of strength for ships above 600 ft. in the International Load Line Rules. To overcome these omissions the straight line extrapolation where $f = \cdot 03L - 2\cdot 7$, already adopted for large existing vessels, was used for lake ships. For sea-going ships the Society's tentative proposal for minimum modulus for large ships was used in association with $C_b = \cdot 86$ and draught $= \cdot 06L$ which resolved into a straight line extrapolation of the published f values giving $f = \cdot 06L - 14$.

In determining the equivalent lake values which are plotted in Fig. 3, three aspects of the problem were included as follows: —

1. Lake f values are in association with fresh water draughts and were reduced by 35/36.
2. Large lakers are permitted by their load line rules to have length to depth ratios of up to 19. To take account of this slenderness the sea-going length to depth ratio was taken as 15 and the lake f value reduced directly for these ratios where applicable.
3. The minimum section modulus of the lake ship, for classification, is a constant proportion above fBd and it was assumed that an equal constant, which is actually variable in ore carriers, is used for the sea-going minimum.

The resultant curve of lake/ocean ratios may not be the perfect representation of relative strength and it might appear on first sight that the reduction in wave stress in Fig. 3 is too great, indicating lack of caution but when it is considered the comparison of strength is wholly in favour of the laker and not the reverse, as might be imagined, it will be seen that caution is an ingredient.

Some experts on this subject cannot accept the statement that longitudinal strength varies directly as draught and the present comparison of f values is contrary to this good opinion. In the present context, however, this is considered permissible on the grounds that if allowance is made for the lakers' diminutive draughts, their equivalent f values would be even further reduced. Also, instead of using a value of 15, item 2 above might well have been related to the length to depth ratio of 13·5 stated in the International Load Line Rules, resulting in a reduced equivalent f value and item 3 can favour the laker by some 4 per cent to 12 per cent. The underlying reasons for item 2 may or may not receive general agreement and the Author will be interested in his colleagues' views which will no doubt come out in the discussion.

Experience of lakers ceases at 700 ft. in length but with the new mammoth lock at Sault Ste. Marie presently under construction, ships in excess of 900 ft. in length may be possible. 850 ft. is seriously being considered at the moment (Ref. 1) and maximum draught, which will be a factor in strength cannot be more than ·03L in association with a length to depth ratio possibly in excess of 20. Correlating such ships to a comparable sea-going standard would appear to be outside the realm of applied mathematics and it is considered that the equivalent lake f values for lengths above 700 ft. in Fig. 3 are at least optimistic.

The point of this is to indicate that the reduction in wave stress shown in Fig. 3 errs on the side of conservatism and consequently so also does the increase to the still water stress. For had the equivalent lake values been lower the permissible lake stress in still water would naturally be higher.

It may be well to comment here that the disparity in strength shown in Fig. 3 in no way implies that the lake ship is unsuitable for lake service; it merely emphasises that she is entirely unfit for the open sea. Theoretical naval architecture should not be divorced from the experience of successful and unsuccessful ships which must always be the fundamental basis for decision and there can be no doubt that large lake ships come within the category of successful experience.

With indeterminate problems the modern tendency is to select an acceptable probability. Higher stresses than the permissible total and still water values shown in Fig. 4 could possibly be sustained without damage to the structure but the probability of damage would obviously be increased. There is no fine, clear line between success and failure and it would be to the benefit of operators if stresses much lower than those shown are obtained in service. Vessels which operate with very low or zero still water stresses have the greatest expectation of being structurally trouble free.

It is also safe to say that the structurally perfect ship has yet to be built. Inferior detail design, erection procedures and workmanship combine to produce discontinuities and stress raising arrangements, particularly in welded construction, which add an unknown quantity in terms of stress and this often produces failure where theoretically none should arise.

The structural arrangements at hatch corners in large lakers are not comparable with the efficient corners now being recommended and fitted in sea-going ships. From the records of heavy weather damage on the lakes, and as already mentioned, the corners are vulnerable when cross seas produce twisting of the hull. In assessing the proportion of total stress to be reserved for wave stress these hatchway corners are of major importance and could have more bearing on success or failure than the traditional critically dimensioned wave. Relatively short waves can produce appreciable twisting of the hull. In determining the permissible still water stress in lakers therefore an empirical factor of 2/3 was introduced as the probable deficiency ratio for hatch corners and the full allowance for reduced wave stress was multiplied by this fraction before being added to the sea-going stress.

Summary of Permissible Stresses in Large Lakers

The curves of stress in Fig. 4 may be expressed as follows:

Still Water Stress

$$=\frac{L}{102} \text{ for L between 400 ft. and 600 ft.}$$

$$=\frac{L}{380}+4\cdot 3 \text{ for L between 600 ft. and 900 ft.}$$

Total Stress

$$=\frac{L}{167}+5\cdot 7 \text{ for L between 400 ft. and 600 ft.}$$

$$=\frac{L}{1000}+8\cdot 7 \text{ for L between 600 ft. and 900 ft.}$$

In the many miscellaneous conditions, with or without ballast, peculiar to a laker's service when the vessel is tied to a quay wall or in the specially sheltered confines of a harbour, still water stresses approaching the total stress value would be permissible.

It is recommended, however, that when operating in the open lakes the still water stresses given above should not be exceeded.

Past storms on the lakes are graded by the violence, destruction and death left in their wakes and each year the month of November can be relied on to produce the greatest storms and damage with October following in second place. During the latter part of the operational season therefore, under the influence of this compelling evidence, every effort should be made to reduce still water stresses to the least possible.

Comparison with Existing Published Data

Over 30 years ago William Bennett investigated the longitudinal strength of a 600-ft. bulk freighter by the classical load and buoyancy curve method and sufficient particulars are available in Mr. Bennett's paper (Ref. 12) to compare the longitudinal stresses by the mean moment method.

The ship he examined was longitudinally framed at deck and bottom and was generally similar to her present-day counterpart except that she was completely riveted and the wing tank longitudinal bulkheads stopped at the main deck. To-day, ships are almost completely welded and the longitudinal bulkheads invariably extend to the spar deck.

His examination was confined to six conditions, of which the first three were in still water and are ideally suited for comparison. Conditions 4, 5 and 6 were taken on a standard trochoidal wave profile, 300 ft. in length by 20 ft. high, with trough at the bow, midships and stern in each case. The cargo loading in No. 4 is a practical operating condition while Nos. 5 and 6 would never be met with in service.

It is intended that only conditions of loading and ballasting actually used by the operators of these ships be examined

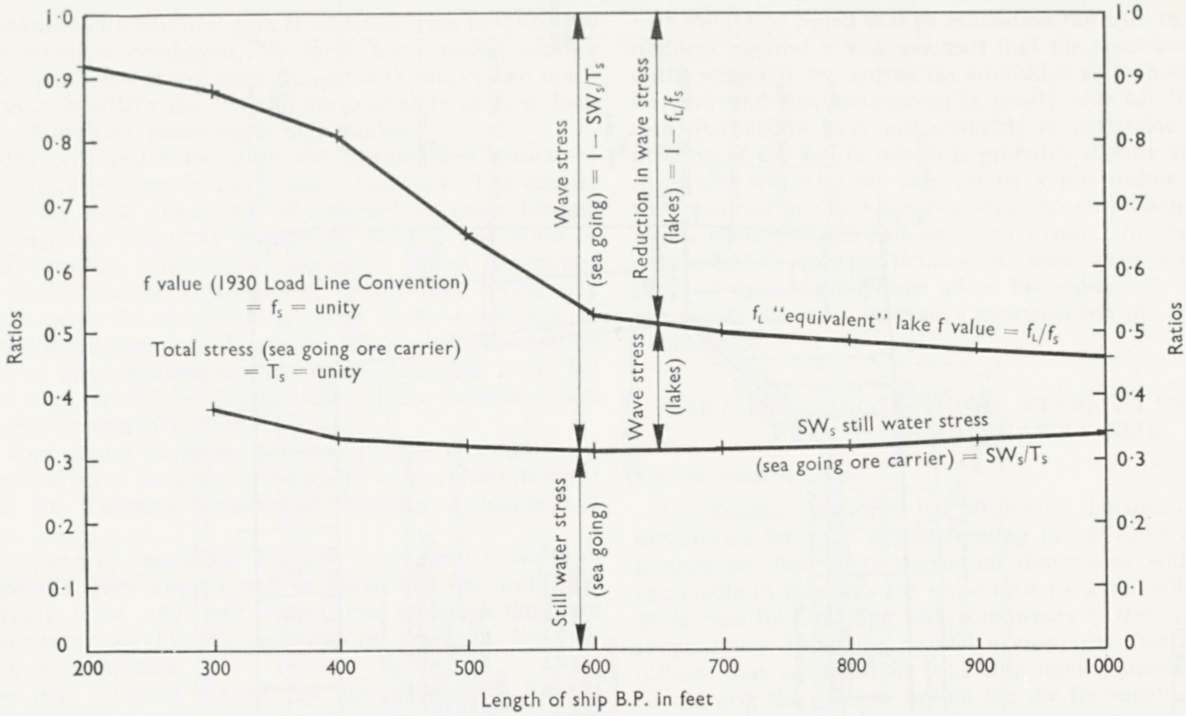


FIG. 3

Ratios of load line f values and stresses

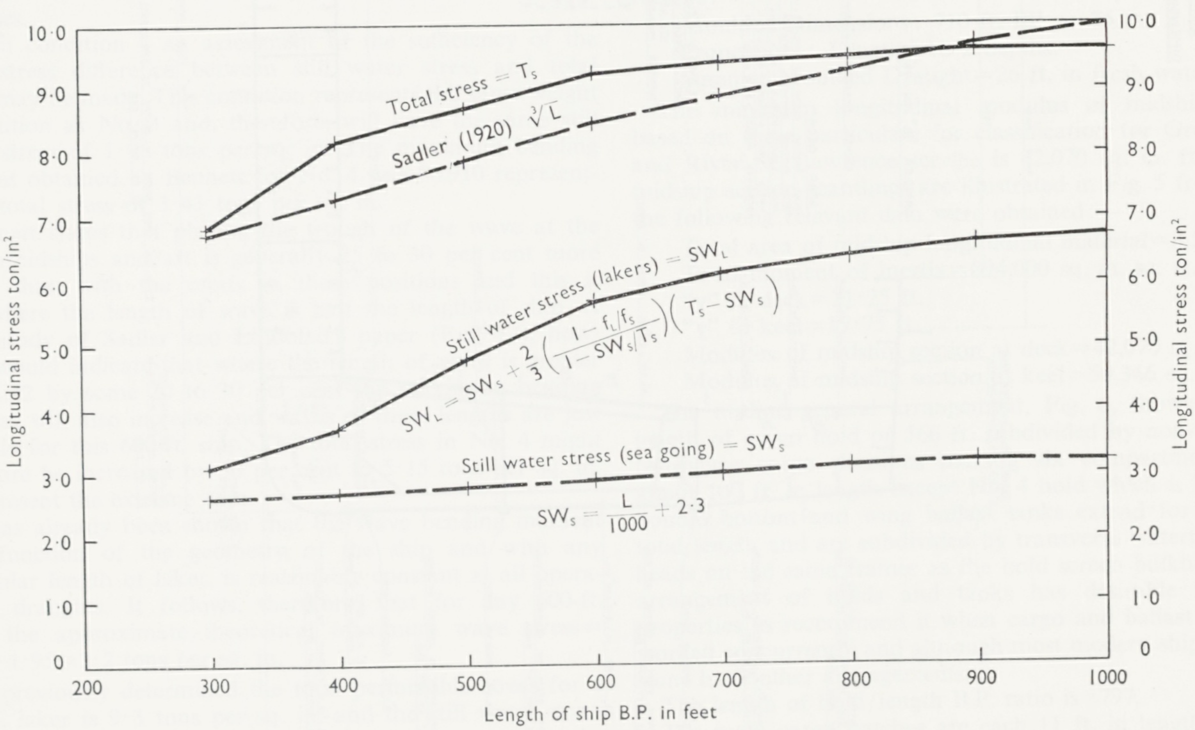


FIG. 4

Permissible total and still water stresses

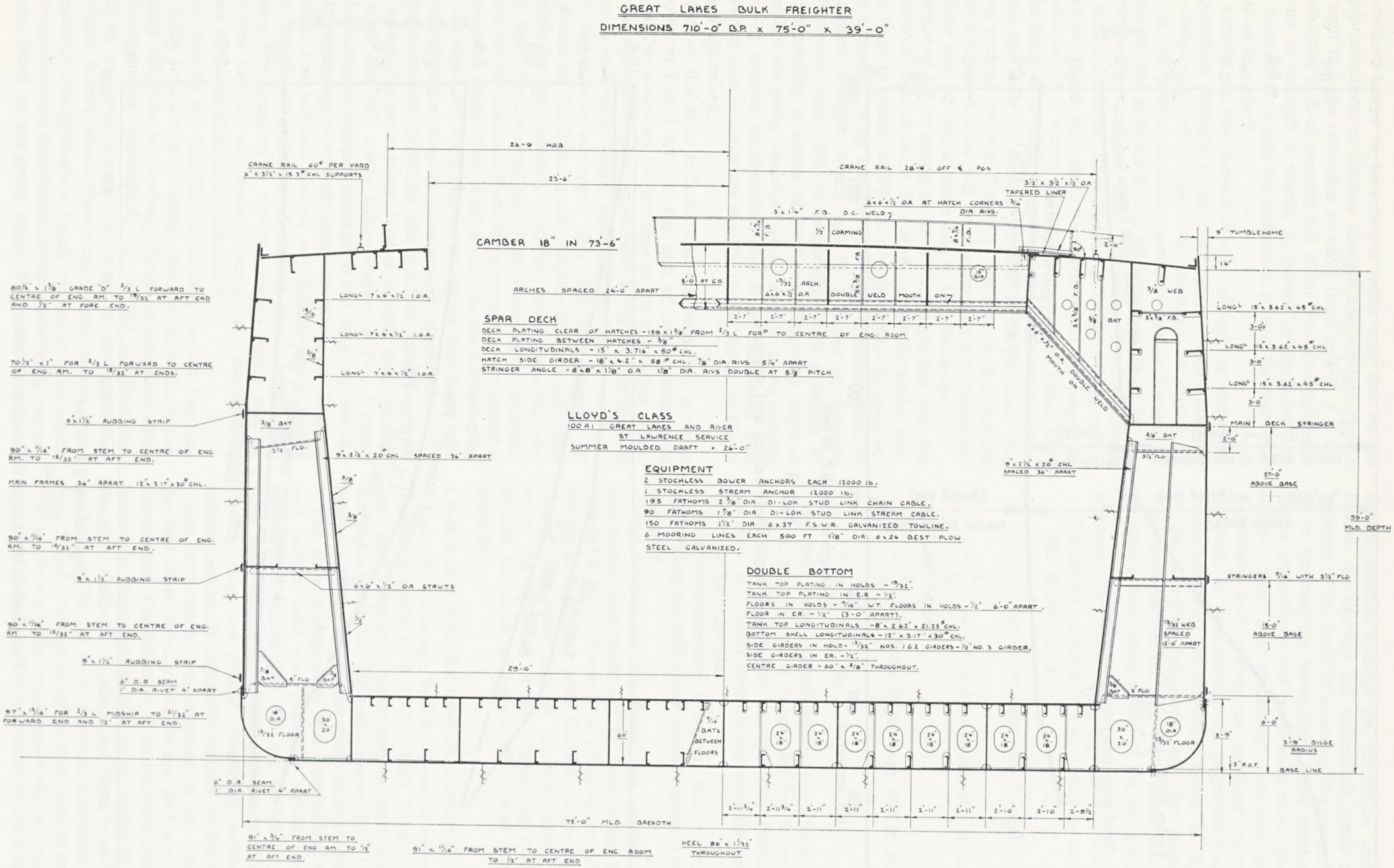


FIG. 5

Midship section

in this paper as no practical gain is achieved from non-realistic and hypothetical conditions. No large laker would venture into the open lakes at her load draught with one or two holds empty even in a flat calm. To add the possibility of 20 ft. high waves and prohibit ballasting is unthinkable.

Conditions 5 and 6 therefore lack realism and cannot be considered as relevant to the present discussion. The nearest approach to these conditions is obtained at grain loading docks when the ship may require to move a few miles to complete loading. When this is necessary ballast is retained in the double bottom tanks in way of any hold which may have no cargo, in an effort to retain some semblance of uniform loading along the length of the ship but the essential features of this operation are that the load draught is not yet attained and the ship is in still water. These conditions are thoroughly examined in Part III.

For comparison purposes therefore Bennett's conditions 1 to 4 provide some measure of the merits of the mean moment method for assessing longitudinal bending moments and stresses.

Comparison of conditions 1, 2 and 3 are given in detail in the appendix from which it will be noted that the results are remarkably close and well within the practical limits of approximation set by the assumptions made in the two methods of calculation.

Some may consider the 15 per cent divergence for the maximum bending moment in condition 3 is not close enough. Bennett's maximum moment of 49,540 gave a stress of 1·95 tons per sq. in. while the mean moment method's SWBM of 42,000 gives a corresponding stress of 1·65 and possibly the difference of 0·3 tons per sq. in. is easier to accept. It is axiomatic that since the SWBM is the difference of two large moments, the mean moment method tends to be less accurate when SWBM is small and increases in accuracy as SWBM increases.

From condition 4 an assessment of the sufficiency of the wave stress difference between still water stress and total stress may be made. This condition represents the same weight distribution as No. 3 and, therefore, will have the same still water stress of 1·95 tons per sq. in. The maximum bending moment obtained by Bennett for No. 4 was 86,910 representing a total stress of 3·43 tons per sq. in.

Bennett states that placing the trough of the wave at the bow, amidships and aft is generally 25 to 30 per cent more severe than with the crests in these positions and this is true where the length of wave is half the length of ship. A close study of Sadler and Lindblad's paper (Ref. 13), however, would indicate that where the length of wave is greater than L/2 by some 20 to 30 per cent the maximum bending moment will also increase and waves of these lengths are just possible for this 600-ft. ship. The total stress in No. 4 might therefore be increased by 50 per cent to 5·15 tons per sq. in. to represent the extreme case.

It has already been shown that the wave bending moment is a function of the geometry of the ship and with any particular length of laker, is reasonably constant at all operational draughts. It follows, therefore, that for any 600-ft. laker the approximate theoretical maximum wave stress= 5·15−1·95=3·2 tons per sq. in.

As previously determined the total permissible stress for a 600-ft. laker is 9·3 tons per sq. in. and the still water stress is 5·8. The balance of 3·5 tons per sq. in., representing the wave stress, is sufficient to accommodate the theoretical maximum as derived above.

It should be noted that in estimating the trim for the mean moment method it was assumed that the combined hull and outfit weight is so centred on amidships as to have no effect on trim and this assumption is nearly true for this type of ship. Particulars were not available to determine the actual position of the L.C.G. which is probably slightly aft of amidships and will trim the ship aft by a few inches more than was obtained in the Appendix. Also, no correction was made to the mean draught in Conditions 1 and 2 for the positions of the L.C.F. since particulars for these were not available. Both of these assumptions affect the values of "a" used in computing the mean buoyancy moments but the amounts are infinitesimal.

PART III—LONGITUDINAL STRESSES IN BULK FREIGHTER OF SEAWAY SIZE

The Subject Vessel

The Seaway Authority has effectually dictated restrictional dimensions for lake ships operating below Lake Erie and it is apparent that these maximum dimensions which will be applicable in this area for some time to come, will be shown preference by Canadian lake shipowners in their replacement programmes. From the Canadian viewpoint then it naturally follows that examination of a ship having these dimensions should give the greatest benefit for the foreseeable future.

For the purpose of this investigation, therefore, and to eliminate the flavour of preference for any particular section of the industry a maximum sized bulk freighter was created from the numerous Canadian designs available. Although not identical to any yet built she bears a striking resemblance to many now afloat.

The essential particulars of this representative laker are as follows:—

Moulded Dimensions: 710 ft. BP × 75 ft. × 39 ft.
Proportions: Length=18·2 depths.
Summer Moulded Draught=26 ft. in fresh water.

The minimum longitudinal modulus of midship section based on these particulars for classification for Great Lakes and River St. Lawrence service is 42,070 sq. in. ft. and her midship section scantlings are illustrated in Fig. 5 from which the following relevant data were obtained:—

Total area of midship longitudinal material=3,550 sq. in.
Total moment of inertia=894,000 sq. in. sq. ft.
"y" to deck=21·25 ft.
"y" to keel=17·75 ft.
Modulus of midship section at deck=42,070 sq. in. ft.
Modulus of midship section at keel=50,366 sq. in. ft.

The outline general arrangement, Fig. 6, shows the total length of cargo hold of 566 ft. subdivided by non-watertight transverse cargo divisions making six compartments, each about 100 ft. in length except No. 4 hold which is 72 ft. The double bottom and wing ballast tanks extend for the same total length and are subdivided by transverse watertight bulkheads on the same frames as the hold screen bulkheads. This arrangement of holds and tanks has desirable functional properties to recommend it when cargo and ballast are being worked concurrently and although most modern ships comply, some have other arrangements.

The length of hold/length B.P. ratio is ·797.

The main cargo hatches are each 11 ft. in length at 24 ft. centres making 22 in all which gives good cargo distribution in the ore trade. This arrangement, which is illustrated in

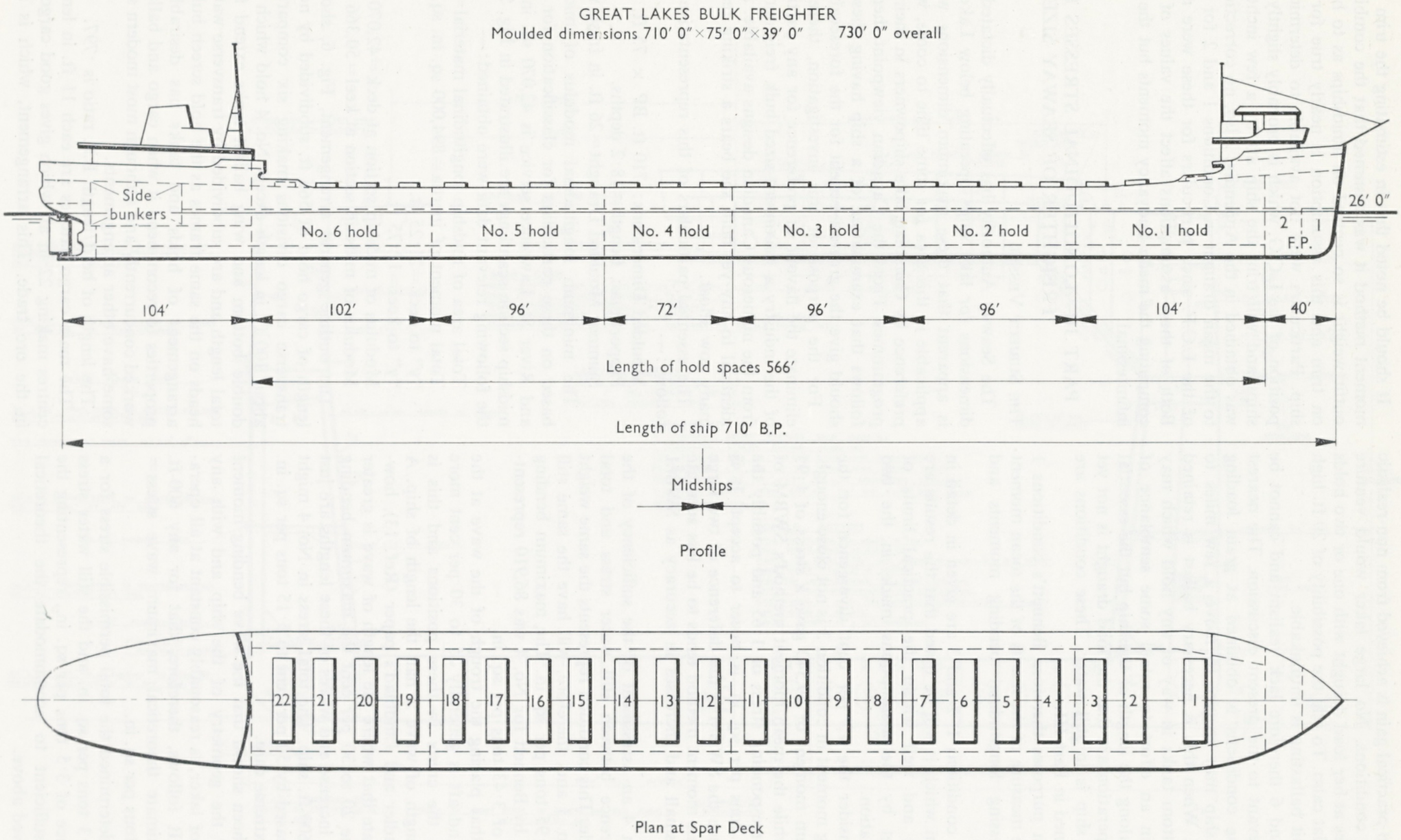


FIG. 6
Outline general arrangement

Plate 1, was universal but is being outmoded and hatches up to 20 ft. in length, shown in Plates 2 and 3, are being fitted with reduced area between the openings. This development facilitates grab discharge but adds to the problem of trim when loading ore with modern mobile belt loaders. The longitudinal positions of the ore piles in the hold become even less predictable than at present and trim can be considerably affected so that topping off cargo is slower. Having regard to the mechanical handling and stowing of steel or aluminium hatch covers and to the existing ore bins at the head of the lakes with spouts fixed at 12 ft. centres it would appear that hatches shorter than 20 ft. in length will eventually become standard. To accommodate these hinged ore chutes hatches must be as wide (transversely) as possible and the practice in large lakers of having the side coaming fitted about 3 ft. 3 in. outboard of the line of hatchway openings allows the chute to lie on the edge of the spar deck opening with the coaming well under the chute. This artifice eliminates the coaming as a potential restriction to depth of ship.

Weight Distribution

The unit of weight used throughout is the "long" ton of 2,240 lb.

The light ship weight was taken as 7,450 tons which is the sum of a hull and outfit weight of 6,825 tons and a machinery weight of 625 tons and is in fair agreement with weights supplied by builders for steam turbine driven vessels. The L.C.G. of the hull and outfit weight was taken at midships and the machinery weight centre at 284·2 ft. aft of midships.

From particulars made available for ships now afloat, the L.C.G. of the light ship excluding machinery, varied between midships and about 5 ft. aft and the adoption of the midship L.C.G. simplified the trim calculation. Some existing large lakers would possibly trim up to about 6 in. more by the stern than the subject vessel for the same ballasted or loaded condition.

Three standard conditions were taken for capacities of fuel, oil, water, etc., namely, full, departure and arrival, and particulars of these are given in Table 2. The "all tanks full" condition represents average maximum capacity with steam turbine installations having two oil fired water tube boilers. Departure was taken as the half capacity condition. The arrival or minimum condition was obtained from the records of ships in operation and represents average practice. Oiling stations servicing lakers are numerous and good estimates of probable fuel consumptions can be made so that the large laker operates on a smaller reserve than sea-going ships and normally carries considerably less than her total capacity of consumable items.

For the loaded conditions investigated in Groups 3 and 4, Table 11, actual capacities of fuel, oil, water, etc., were taken from the ships' records.

The maximum capacities and longitudinal centres of gravity for the water ballast tanks are shown in Table 3 and are based on fresh water content. It should be stated here that the L.C.G. for tanks when full was used when computing the moments for tanks partially filled and no correction was made for trim. The error in the longitudinal stresses resulting from this omission is small and considered acceptable.

The weights and moments forward and aft of midships for fresh water ballast in holds, Group 2, Table 10, were assessed taking final trim into account, that is, the surface of the water in the holds was parallel to the ship's water line with the ship in equilibrium, and were computed for various depths of water on the engine room forward bulkhead. The depths used for each condition in Group 2 are tabulated in Table 5 and the weights of ballast in holds in Table 10.

The cargo weights in Groups 3 and 4, Table 11, were taken from the records of actual ships and required slight modification to fit the subject vessel. Complete details of weights with a description of the manner and sequence of loading are given later.

Draughts and Trim

Draught and trim are extremely important factors in lake ship operation whether she is loaded or in ballast and in order that the practical aspect of the conditions investigated might be better appreciated, particulars of these were calculated from the hydrostatic curves reproduced in Fig. 7. Draughts aft and forward are normally of more importance than

TABLE 2

FUEL, OIL, WATER, ETC., CAPACITIES

ITEM	WEIGHT (Tons)			L.C.G. (ft.)
	FULL	DEPARTURE	ARRIVAL	
Oil Fuel	830	415	80	320·0a
Diesel Oil	14	7	2	250·0a
Lub. Oil	15	8	3	278·0a
Feed, Distd., San. Water	250	125	30	293·0a
Drinking Water	40	20	5	320·0f
Stores and Crew Aft	15	10	5	280·0a
,, ,, ,, Fwd.	15	10	5	320·0f
Totals (Tons)	1179	595	130	—

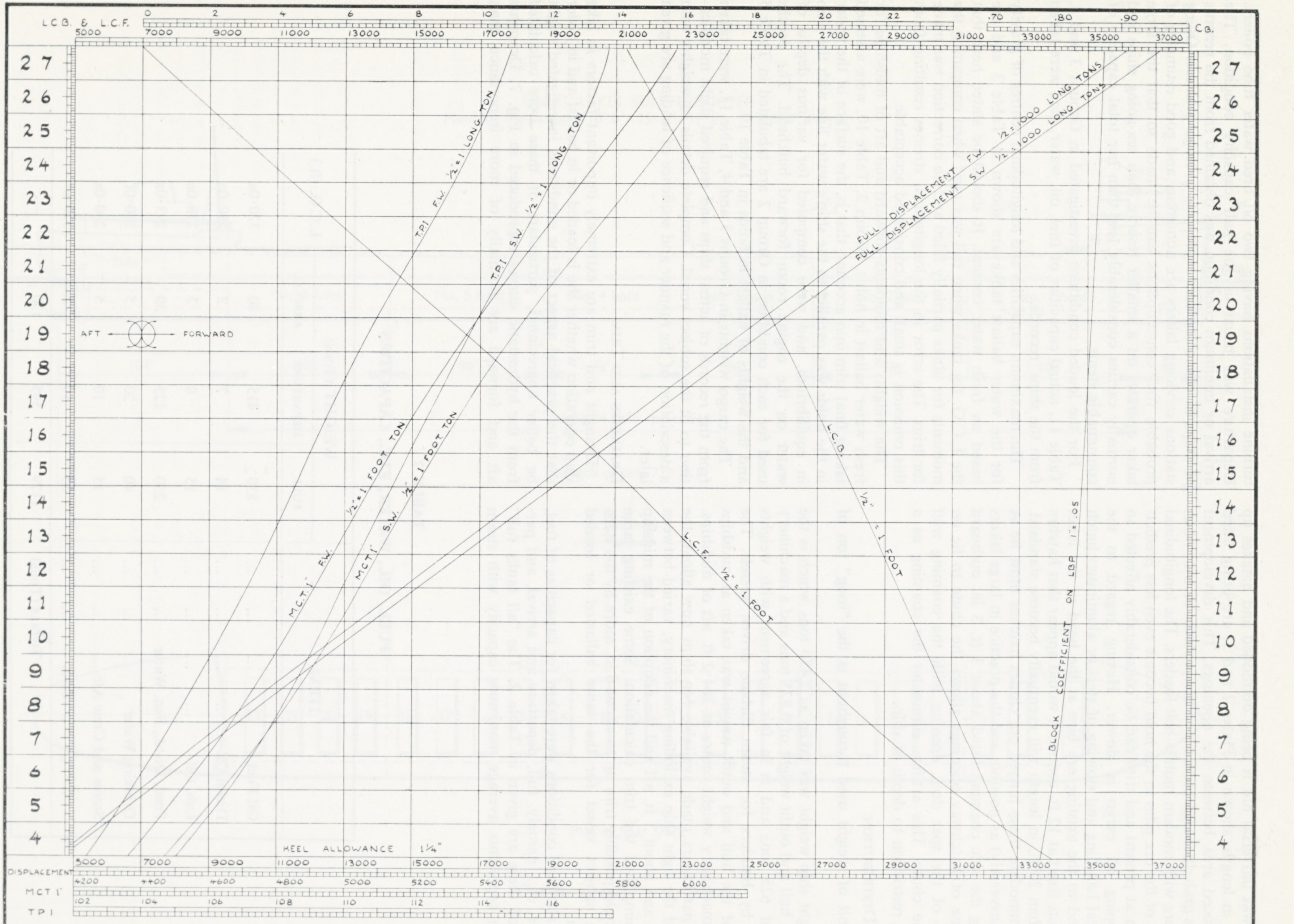


FIG. 7
Hydrostatic curves for 710 ft. × 75 ft. × 39 ft. laker

TABLE 3

BALLAST TANK CAPACITIES—FRESH WATER

COMPARTMENT	CAPACITY (Tons)	L.C.G. (Feet)
No. 1 Fore Peak Tank	210	343·0f
No. 2 ,, ,, ,, ...	650	323·4f
No. 1 D.B. and Wing Tank	1852	256·3f
No. 2 ,, ,, ,, ,, ...	1900	167·9f
No. 3 ,, ,, ,, ,, ...	1900	64·8f
No. 4 ,, ,, ,, ,, ...	1400 { 319 / 1081 }	8·2f / 27·8a
No. 5 ,, ,, ,, ,, ...	1900	103·7a
No. 6 ,, ,, ,, ,, ...	1900	200·6a
TOTAL (Tons)	11712	—

midship draught and all three are, therefore, included in Tables 10, 11 and 12 for each condition examined.

Ballasting and loading are almost entirely governed by the draughts aft, forward and amidships and in their simplest form the controlling requirements may be stated thus:—

1. Draughts aft and forward must not exceed the mandatory or recommended maximum draught for the canals and restricted areas included in the voyage.
2. Draught aft must always cover the propeller (20 ft. 4 in. in the case of the subject vessel).
3. Draught forward in ballast should be at least 3 ft. and may be increased to anything up to about 18 ft. depending on wind and weather.
4. The statutory seasonal load line mark at midships, applicable for the voyage, must not be submerged.

Ships on the whole operate in accordance with the foregoing although 2 and 3 are at the Masters' discretion. In the Welland and St. Lawrence Canals 1 is enforced but in other areas where maximum draughts are recommended some small latitude can be expected and this tends to take the form of sag in which the midship draught, and consequently deadweight, is increased but the increase is small. Large lakers built in the last few years have draughts at least equal to those corresponding to the load line markings shown in Fig. 1 and because of the draught restrictions in the Seaway and Sault Ste. Marie channel, 4 above never applies in the midsummer and summer seasons and only sometimes in the intermediate season. This limits requirement 4 to the winter season which is a very small part of the total annual operational period.

The ships are never trimmed by the head in operational ballast conditions and rarely when loaded. Ballast suctions are only fitted at the aft end of tanks and hold and this practice has never been found inconvenient when flooding or draining.

Successful operation has been accomplished in the past, and presumably will continue in the future, on considerations of draught requirements without reference to loading scales or diagrams and it is most important this basic fact be recognised, and one's acclimatisation to it be complete, before proceeding to consider the longitudinal stresses such action induces in service.

PREAMBLE TO INVESTIGATION

It is intended to portray the complete range of conditions in so far as they affect longitudinal stresses, from the minimally light ship to the completely loaded and completely ballasted conditions, and this intent has entailed the examination of 124 separate conditions from which a reasonable picture can be drawn.

These ships are capable of an infinite variety of conditions some of which could well be detrimental to the hull structure but personal contact with masters and deck officers has produced a firm conviction that these gentlemen are aware of the structural limitations of their charges and that they act with conservatism well within the bounds of safe and efficient handling. They are not so well aware of the theoretical and technical extent of these limitations and theirs is action based on tradition and convention with little assistance from theory on how far deviations from their code of practice can be taken before harm is done. Thus the tendency is to err on the side of safety and this too may be accentuated in their minds by the prodigious cost of their ships together with the fact that jobs are not too easy to come by.

As a digression from the main theme of this paper, it should be recorded, with complete sincerity, that in the Author's experience, senior officers on Canadian-owned large lakers are practical and competent men and can have few equals to the quiet efficiency they exhibit in all phases of operation. Their tender but active handling of such immense ships in Seaway locks is an object lesson in the art of applied experience demanding patience, tolerance and fortitude, all of which are demonstrated in such casual manner as to give the erroneous impression that there is nothing to it. Such are the men who operate these ships.

To continue, it is considered that the conditions investigated adequately and, more to the point, realistically cover, within the confines of practical seamanship, the stresses induced in the every day operation of large lakers. For convenience they are presented under three main headings which are further subdivided into a total of seven groups.

Summary of Conditions Examined

Operational Ballast Conditions, Table 10

Group 1 *Condition Nos*. 1 *to* 17

Varying amounts of water ballast in double bottom and wing tanks only, Table 4, are considered with no water ballast in holds and no cargo.

Group 2 *Condition Nos*. 18 *to* 61

Varying amounts of water ballast in holds, Table 5, are considered with the water ballast in tanks represented by Condition Nos. 8, 10 and 12 to 17: no cargo in holds.

Loaded Conditions, Table 11

Group 3 *Condition Nos*. 62 *to* 74

Nos. 62 to 71 represent the actual loading of a laker very similar to the subject vessel on 12th August, 1961, at Seven Islands ore dock.

No. 62 is the light condition immediately before loading commenced.

Nos. 63 to 70 are partially loaded conditions at intervals of 3,000 tons of cargo on board.

No. 71 is the final topped off and departure condition in salt water at Seven Islands.

No. 72 is the actual arrival condition at St. Lambert Lock for entry to the St. Lawrence Seaway in fresh water.

No. 73 is the arrival condition at Cleveland with a small amount of ballast to maintain near level trim for docking.

No. 74 is an actual departure condition on 24th August, 1961, at Port Arthur trimmed for the Sault Ste. Marie channel draught, loaded with grain for Quebec with the cargo self trimmed and all hatches full.

Group 4 *Condition Nos*. 75 *to* 83

This group illustrates the mode of working cargo and ballast together and is taken from the records of a ship which arrived in ballast condition No. 15 on 6th October, 1961, at Port Arthur and left two days later loaded with grain in condition No. 83.

Miscellaneous Conditions, Table 12

Group 5 *Condition Nos*. 84 *to* 87

These are light ship conditions with no water ballast and no cargo but with the varying amounts of oil fuel, fresh water, etc., shown in Table 2.

No. 84 is the minimum light ship with no oil fuel, fresh water, crew, stores, etc.

No. 85 is the minimum operational arrival condition.

No. 86 is the half capacity condition.

No. 87 is with maximum capacity of oil fuel, fresh water, etc.

Group 6 *Condition Nos*. 88 *to* 110

These are transition conditions through which the large laker would take on ballast when in light ship condition No. 86 to attain any of the operational ballast conditions in Group 1. They can, of course, be viewed in reverse in which any ballast condition in Group 1 would be reduced to the light ship condition No. 86.

Group 7 *Condition Nos*. 111 *to* 124

This group illustrates the many conditions possible when tipping the vessel for propeller examination afloat.

Operational Ballast Conditions

Condition Nos. 1 to 61, Table 10, are derived from the actual experiences of a large laker so far as draughts and quantities are concerned and cover the operational ballast conditions from the approximate minimum, No. 1, with 5,630 tons, to the maximum, No. 47, with 18,056 tons when hold flooding is included. The range is representative of the general code of practice adopted by officers in lake service and also deviates sufficiently to embrace the practical extremes met with in operation.

The departure capacities for fuel oil, fresh water, etc., Table 2, are used throughout.

For convenience in identification, Group 1, Nos. 1 to 17, has no water ballast in holds and Group 2, Nos. 18 to 61, is with water ballast in holds but there is no actual division in practice. Hold flooding could be commenced any time after the ballast tanks contain about 7,000 to 8,000 tons of water and also the condition with maximum ballast tank capacity, No. 17, could be used without hold flooding.

The amount of water ballast taken on board for operational voyages broadly falls into two classes, one for navigation in the open lakes and the other for negotiating locks and approach channels. All of the conditions in Groups 1 and 2 could be used in the open lakes although in special circumstances some are infinitely better than others. For navigation through the Seaway and depending on weather, a draught forward of at least 10 ft. would be necessary with the propeller immersed aft.

For normal running in the open lakes in very calm weather and where maximum speed might be a factor, the least possible ballast would be taken, consistent with keeping the propeller immersed and with about 3 ft. draught forward. This is the condition in which the highest speeds for individual vessels are recorded and is represented by No. 1, Table 10. From Tables 3 and 4 it will be seen that Nos. 4 and 5 ballast tanks are full and the ship trimmed on Nos. 3 and 6 tanks which are partially full.

As wind increases and weather deteriorates ballast is increased and to accomplish this while maintaining minimal propeller immersion the sequence Nos. 1, 3, 5, 9, 11 and 13, Table 4, would in general be used by the more astute officers and gives best draughts for the least ballast.

An alternative of following sequence Nos. 2, 4, 6, 10, 12 and 13 is sometimes adopted in which Nos. 4, 5 and 6 tanks are completely filled and trim is reduced by progressively filling Nos. 3, 2 and 1 tanks. Examination of the resultant draughts and amounts of ballast necessary, shows the former sequence to give lower stresses than the latter indicating better weight distributions but as will be seen from Fig. 8 the differences in stress are slight.

Nos. 7 and 8 are included to illustrate the increase in stress by commencing to fill No. 1 tank before No. 2 tank is full.

Up to this point, No. 13, with about 10,000 tons of water ballast aboard, that is, Nos. 2 to 6 tanks full and No. 1 tank 50 per cent full, the still water stresses are low and, having regard to trim and draught requirements, conditions detrimental to the hull structure cannot be produced. Also for normal operation hold flooding would possibly have commenced.

Nos. 14, 15, 16 and 17 would rarely be used in the open lakes without hold flooding as the draughts aft are becoming too small and in addition, with their slow rate of filling and discharge, the two fore peak tanks are seldom used. These conditions would be used, however, when going to load grain at the head of the lakes where a mean draught of 15 ft. to 16 ft. may be necessary to get under the spouts. It can take up to six hours to clean the holds after flooding and this is, therefore, avoided before grain except in exceptionally bad

TABLE 4

BALLAST IN TANKS, GROUPS 1 AND 2
OPERATIONAL BALLAST CONDITIONS

Condition	Double Bottom and Wing Tanks						Fore Peaks		Total Ballast (tons)
	6	5	4	3	2	1	2	1	
1	1100	1900	1400	1230	—	—	—	—	5630
2	1900	1900	1400	1230	—	—	—	—	6430
3	1030	1900	1400	1900	—	—	—	—	6230
4	1900	1900	1400	1900	—	—	—	—	7100
5	1250	1900	1400	1900	950	—	—	—	7400
6	1900	1900	1400	1900	950	—	—	—	8050
7	1525	1900	1400	1900	950	463	—	—	8138
8, 18, 19	1900	1900	1400	1900	950	463	—	—	8513
9	1475	1900	1400	1900	1900	—	—	—	8575
10 & 20 to 23	1900	1900	1400	1900	1900	—	—	—	9000
11	1750	1900	1400	1900	1900	463	—	—	9313
12 & 24 to 27	1900	1900	1400	1900	1900	463	—	—	9463
13 & 28 to 32	1900	1900	1400	1900	1900	926	—	—	9926
14 & 33 to 37	1900	1900	1400	1900	1900	1389	—	—	10389
15 & 38 to 47	1900	1900	1400	1900	1900	1852	—	—	10852
16 & 48 to 56	1900	1900	1400	1900	1900	1852	650	—	11502
17 & 57 to 61	1900	1900	1400	1900	1900	1852	650	210	11712

TABLE 5

FRESH WATER BALLAST IN HOLDS, GROUP 2
OPERATIONAL CONDITIONS WITH HOLD FLOODING

Condition	Depth of Water on After Hold Bulkhead
18, 20, 24, 28, 33, 38, 48, 57	2′ 0″
— — — — — — — 58	3′ 0″
19, 21, 25, 29, 34, 39, 49, 59	4′ 0″
— — — — — 50, 60	5′ 0″
— — — — — 51, —	5′ 6″
22, 26, 30, 35, 40, 52, 61	6′ 0″
— — — — — 53	6′ 6″
— — — — — 54	7′ 0″
— — — — — 55	7′ 6″
23, 27, 31, 36, 41, 56	8′ 0″
— — 42	9′ 0″
— — 43	9′ 6″
32, 37, 44	10′ 0″
45	10′ 6″
46	11′ 0″
47	11′ 6″

weather. Under these conditions the highest still water stress of 4·12 tons per sq. in. is acceptable. Incidentally, no cleaning of holds is required between flooding and taking on ore, coal or stones in bulk.

Water ballast is always carried in the holds when negotiating canals in the light condition and also for most of the time when in the open lakes, so that Group 2, Nos. 18 to 61, covers the major part of ballast voyages.

Some of the conditions included in Group 2 are outside actual operational procedure and have been included to give the fringe spots necessary to complete the curves of still water stresses shown in Fig. 9. This applies to Nos. 38, 48, 57, 58 and 59 since in these conditions the propeller is not properly immersed. Nos. 23, 32, 37, 45, 46 and 47 are also unlikely since the draught aft in each case is in excess of the St. Lawrence Seaway maximum. With the wide range of much more suitable draughts available and the versatile ballasting equipment on board it is safe to say that these conditions will not be used in service.

It will be seen from Fig. 9 that the maximum operational still water stress with hold flooding is less than $3\frac{1}{2}$ tons per sq. in. when both fore peak tanks are full and that the stress in normal service is likely to be under $2\frac{1}{2}$ tons per sq. in.

It is interesting to note that when 3 to 4 ft. of water is in the aft end of the hold further additions by flooding reduce the still water stresses.

LOADED CONDITIONS

Loading ore at Seven Islands is carried out by two conveyor belts which are separately mobile on a track on the quay wall parallel to the ship's length and project transversely across the hatchway openings on hinged elevating arms.

Condition Nos. 62 to 71 follow the course of loading from light ship with no ballast to loaded with 24,306 tons of ore on board and Fig. 10 shows the curve of longitudinal stress at deck on a base of cargo added to the ship. All the necessary particulars for computing Nos. 62, 70 and 71 were made available by the owners and all phases of this type of operation have been witnessed by the Author.

The intermediate conditions, Nos. 63 to 69, are good approximations and for the practical purposes of this paper indicate the general trend in this type of loading The amounts of ore in holds for each condition are given in Table 6 and, up to No. 69, are taken at intervals of 3,000 tons of which 1,500 tons have been assumed forward and aft of the bulkhead between Nos. 11 and 12 hatchways. Sometimes delivery by one of the belts is delayed and rarely does this ideal equal distribution about amidships apply but from an instance witnessed in which the forward loader was out of commission for some hours the after loader was so controlled and diverted from the general loading pattern as to indicate that the longitudinal stresses in this emergency would not vary greatly from those shown in Fig. 10. The deck officer's concern for trim and avoidance of premature sag unwittingly operate in favour of the near uniform reduction in hogging stress no matter the circumstances to be surmounted.

The transition stage between arrival at the loading harbour in ballast and the "all ballast tanks" empty condition in which loading commenced is described in Group 6. It is assumed, therefore, that the ship is now tied to the loading quay with

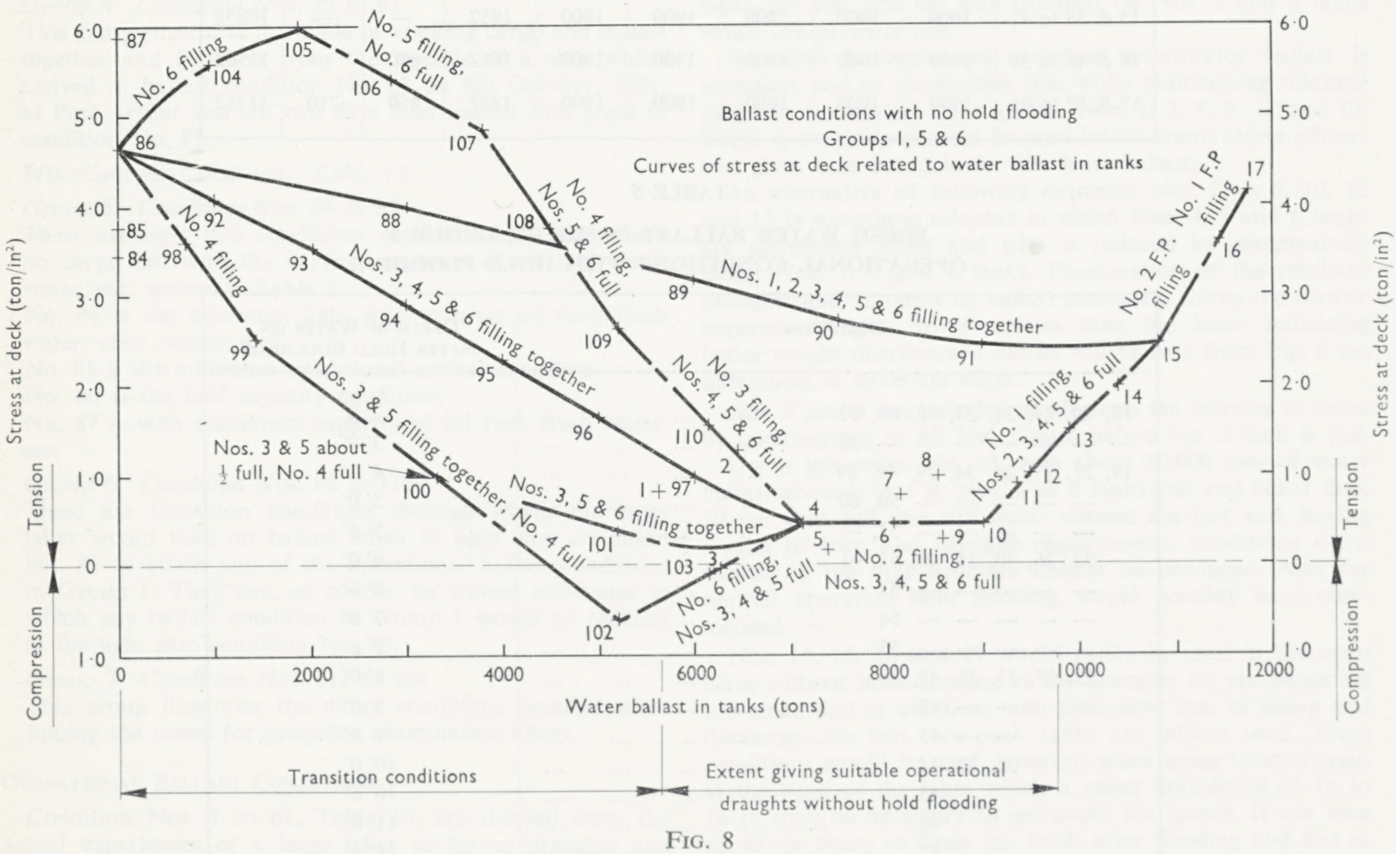


FIG. 8

Ballast stresses with no hold flooding

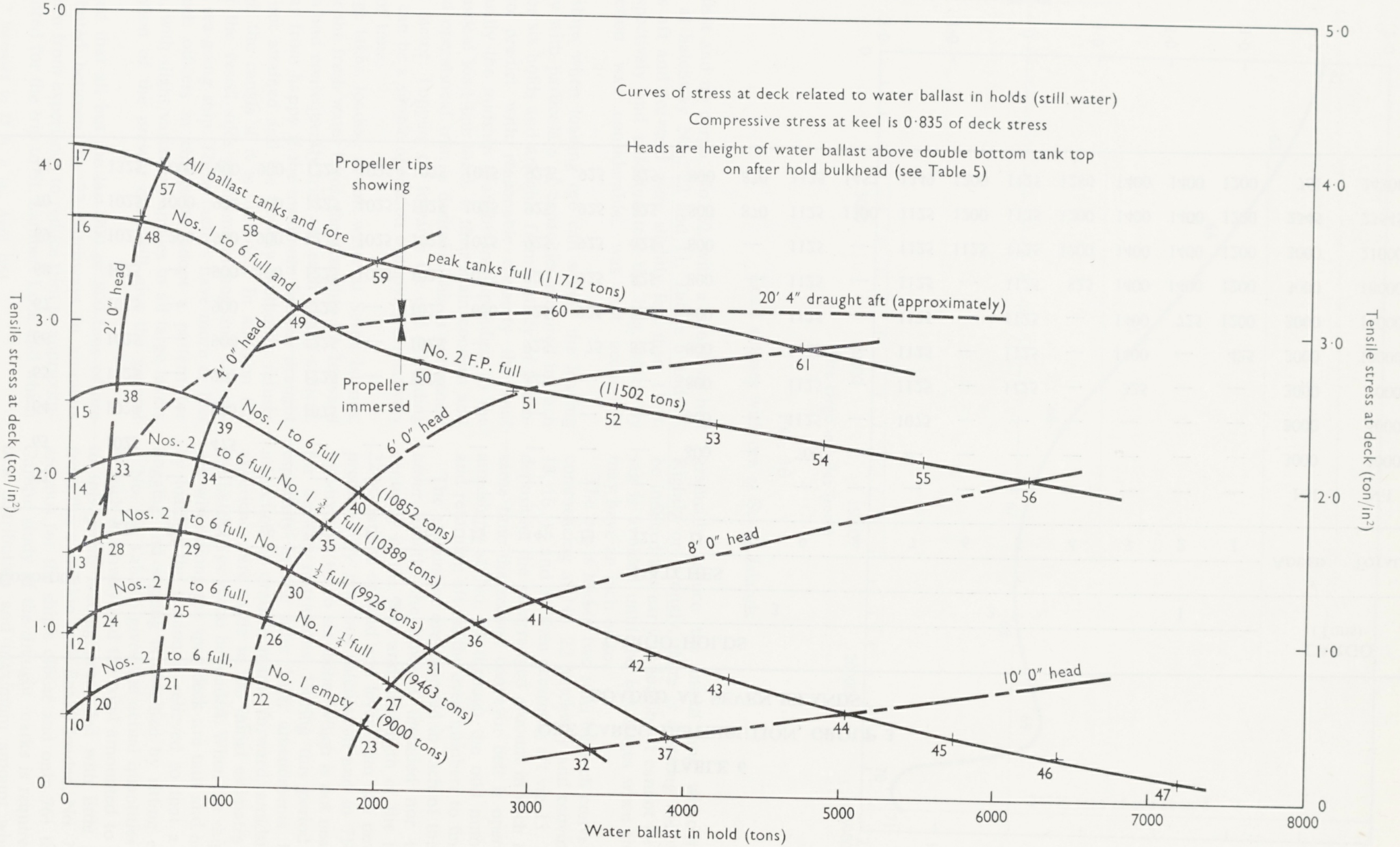


FIG. 9

Ballast stress with hold flooding

TABLE 6

ORE CARGO DISTRIBUTION, GROUP 3

LOADED AT SEVEN ISLANDS

Condition	Cargo Holds																						Cargo (Tons)	
	6				5				4			3				2				1				
	Hatches																						Added	Total
	22	21	20	19	18	17	16	15	14	13	12	11	10	9	8	7	6	5	4	3	2	1		
62	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	Nil	Nil
63	1025	—	475	—	—	—	—	—	—	—	—	800	—	700	—	—	—	—	—	—	—	—	3000	3000
64	1025	—	900	—	1075	—	—	—	—	—	—	800	—	1125	—	1075	—	—	—	—	—	—	3000	6000
65	1025	—	900	—	1225	—	1025	—	325	—	—	800	—	1125	—	1125	—	1125	—	325	—	—	3000	9000
66	1025	—	900	—	1225	—	1025	—	925	75	825	800	—	1125	—	1125	—	1125	—	1400	—	425	3000	12000
67	1025	—	900	—	1225	—	1025	650	925	925	825	800	—	1125	—	1125	—	1125	—	1400	725	1200	3000	15000
68	1025	—	900	100	1225	1025	1025	1025	925	925	825	800	—	1125	—	1125	—	1125	825	1400	1400	1200	3000	18000
69	1025	700	900	900	1225	1025	1025	1025	925	925	825	800	—	1125	—	1125	1125	1125	1200	1400	1400	1200	3000	21000
70	1025	1000	900	900	1225	1025	1025	1025	925	925	825	800	870	1125	1300	1125	1200	1125	1200	1400	1400	1200	2545	23545
71	1325	1000	900	900	1225	1025	1025	1025	925	925	825	800	870	1125	1457	1349	1200	1125	1280	1400	1400	1200	761	24306

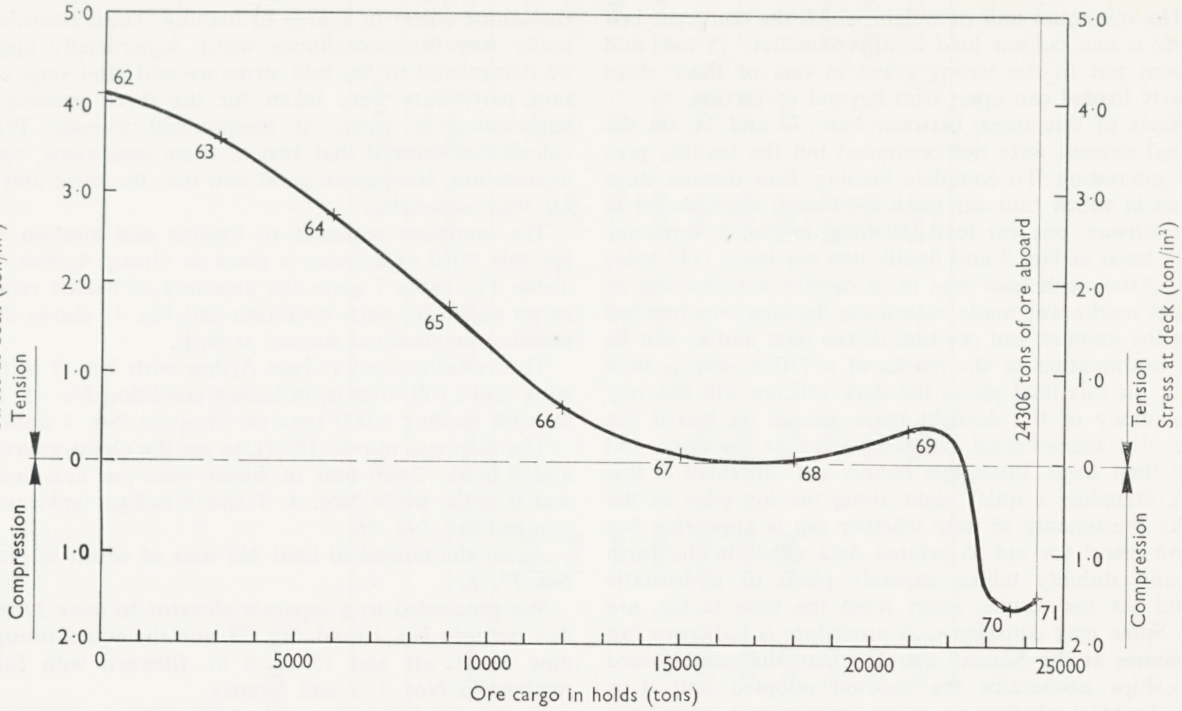


FIG. 10

Load stresses with ore at Seven Islands

no water ballast and no cargo, No. 62, with a conveyor belt at the ready at hatchway Nos. 22 and 11, Tables 6 and 11. The draughts aft and forward were slightly less than 11 ft. and 3 ft. respectively and unfortunately the hog, which by visual inspection was considerable, was not accurately measured.

Sea-going ships when loading ore can supply the loading port authority with particulars of capacities required in each of their five or six holds and are able from a study of hydrostatic data to predict with reasonable certainty the final draughts. Usually the midship draught corresponding to the applicable seasonal load line mark is their main concern and although good operational trim is desirable this is not generally a critical factor. Topping off towards the end of loading for final trim can be a simple operation occupying a relatively short period of time.

For the large laker, loading in salt water at Seven Islands, and bound for the fresh water lakes, the draughts aft, forward and amidships and consequently trim are each vitally critical. Owners are far from happy if the maximum Seaway draught of 25 ft.* is not attained and the Seaway Authority refuses transit through the canals if it is exceeded. In addition the slenderness of the vessel adds problems in hog and sag unknown to the sea-going ship. These factors have more or less forced the deck officers in charge to adopt a pattern for loading which, with slight variation, applies to all large lakers and a description of the present case will give the general picture.

It is arranged that all but the last 700 or 800 tons of ore will be distributed between the 22 hatchways in amounts which are known from experience to reduce trim to approximately that required for the trip to St. Lambert Lock and which gradually reduce the initial hog without producing sag. Rightly or wrongly, there is a strong belief that should sag be induced too early in loading it cannot be corrected and so sag is avoided until the last stages when the final condition may have an inch or so.

The ore is laid in two runs by each conveyor belt with one commencing at No. 22 hatchway and moving forward to 20, 18, 16, 14 and 12 then returning aft via 13, 15, 17, 19 and 21 depositing the prearranged amount in each hatchway. At the same time the second conveyor belt is operating forward of amidships at No. 11 through the odd numbered hatchways and returning aft on the even numbers to finish at No. 10.

The amount of ore dropped in each of the 22 hatchways is never exactly the amount intended nor is each dropped centrally in the fore and aft length of the hatch. Thus with 22 variations of load and lever, trim can never be exactly as predicted by previous calculation and in 710 ft. can be out by anything up to six inches which is not nearly good enough for Seaway restrictions. During this period of loading and especially in the later half, amendments based on visual inspection of draught marks forward, amidships and aft and experience, are made to the initial estimates of the load for the hatchways yet to be loaded. When the ship hogs the guard rail wires along the spar deck are taut and as hog is reduced by loading the tension is relieved so that a rough guide to hogging and sagging is obtained by sitting on the top wire.

No. 70, Table 6 gives the actual quantities of ore dropped in each hatchway and the total amounted to 23,545 tons. Up to this point, loading proceeded with little interruption but from this condition until finally loaded, No. 71, was long and arduous for the ship's officer and only 761 tons were added. Careful study of the draught marks is required to predict the best distribution and maximum amount which can still be

* Increased to 25 ft. 6 in., April 1963.

loaded. The minimum unit of weight which the conveyor belt can deliver is one rail car load of approximately 75 tons and this amount put in the wrong place in one of these ships when nearly loaded can upset trim beyond correction.

The effects of this stage, between Nos. 70 and 71, on the longitudinal stresses were not computed but the loading procedure is interesting. To complete loading, four distinct steps were taken in which four car loads (300 tons) were placed in No. 22 hatchway, one car load (80 tons) in No. 4, three car loads (224 tons) in No. 7 and finally two car loads (147 tons) in No. 8. After each load was in, a careful examination of the draught marks was made before the decision was reached regarding the amount and position of the next and as can be imagined examination of the marks of a 710-ft. ship is time consuming. In this final phase the deck officers will not rely on the accuracy of the draught mark gauges on board but prefer to visit the external painted marks at the ends and middle of their ships. Intangible factors are concealed in this operation, including a quick sight along the top edge of the sheerstrake presumably to note whether sag is appearing but there is no resort for aid to printed data either in the form of trim and stability tables, capacity plans or hydrostatic curves and yet the results, apart from the time factor, are excellent. Some may consider such procedure is ludicrous but the test comes at the Seaway canals where the good record of these ships exonerates the method adopted and it is extremely doubtful whether sea-going practice with its various shortcomings could produce such successful consistency.

The calculated draughts for the completely loaded ship, No. 71, Table 11, agree with these taken before the ship left Seven Islands and there was neither hog nor sag present. The actual draughts on arrival at St. Lambert Lock in fresh water, No. 72, were 24 ft. 11 in., 25 ft. and 24 ft. 11 in. indicating 1 in. sag and at Port Weller for entry to the Welland Ship Canal there was $1\frac{1}{2}$ in. sag due to consumption of fuel oil, water, etc. Before entering Cleveland 100 tons of ballast was taken in No. 6 tank reducing the head trim to 1 in., No. 73, and the sag on arrival was $1\frac{1}{2}$ in.

Apart from the initial stress before loading when in still water, the stresses produced in this entire operation are very moderate and support the contention that these ships are handled with competency.

No. 74, Table 11, is also an actual condition, for the same ship as above, for a cargo of grain loaded at Port Arthur. The sequence of loading was not available but the final cargo consisted of 3,755 tons of barley in No. 1 hold, 3,522 in No. 3, 2,663 in No. 4, 3,411 in No. 5 and 3,659 in No. 6 with 4,374 tons of wheat in No. 2 hold. The ship left with an actual hog of 2 in. and a trim aft of 14 in. which is normal practice where full cubic capacity cargoes are carried as in this case with light weight barley and where mandatory draught requirements permit. Fuel consumption, etc., reduces trim aft and hog so that whenever possible these are initially induced when loading, as opposed to having no trim with some sag, which again confirms the soundness of the operational code of practice.

Loading grain at the head of the lakes, apart altogether from draught and elevator restrictions, is never a straightforward business. Usually more than one type or quality of grain is carried necessitating the vessel moving from one elevator to another in any condition of load or ballast. Through the courtesy of a lake shipowner the Author was privileged to examine the complete records of grain loading for a large laker and to discuss this type of operation with the senior officer in charge of loading. These records revealed many surprising conditions which superficially appeared to be detrimental to the hull structure and after long consideration particulars were taken for the three potentially worst experiences in terms of longitudinal stresses. Preliminary calculation showed that two of these conditions, contrary to expectation, were quite good and that the third and worst of all, was reasonable.

The complete sequence of loading and ejection of ballast for this third experience is given in Group 4, Nos. 75 to 83, Table 11. Table 7 gives the amounts of ballast retained and cargo added for each condition and Fig. 11 shows the corresponding longitudinal stresses at deck.

The vessel arrived at Port Arthur with Nos. 1 to 6 DB and wing tanks full, that is, in ballast condition No. 15, and commenced loading 2,000 tons of wheat in No. 4 hold, No. 75.

The ship was moved 100 ft. to get the elevators over Nos. 5 and 6 holds. 2,680 tons of wheat were put into both Nos. 5 and 6 holds while Nos. 4, 5 and 6 ballast tanks were being pumped out, No. 76.

Again she moved to load 840 tons of wheat in No. 2 hold, No. 77.

She proceeded to a separate elevator to have 1,000 tons of flax put into No. 1 hold, No. 78, and the actual draughts were now 21 ft. aft and 15 ft. 6 in. forward with full ballast retained in Nos. 1, 2 and 3 tanks.

In this condition she moved five miles to have 3,540 tons of wheat in No. 3 hold and 330 tons in No. 4 hold while No. 3 tank was pumped out during loading, No. 79.

The vessel lay all night (eight hours) at actual draughts of 22 ft. aft and 17 ft. 3 in. forward and in the morning moved five miles to have 2,805 tons of flax in No. 1 hold while No. 1 ballast tank was being pumped dry. Nos. 1 and 4 holds were now full and No. 2 ballast tank full, No. 80, and the actual draughts were 21 ft. aft and 20 ft. forward.

She had to move eight miles to have 3,000 tons of wheat in No. 2 hold and 855 tons in No. 3 hold. No. 2 ballast tank was pumped out while loading and the position became Nos. 1, 3 and 4 holds full with no water ballast remaining, No. 81, and actual draughts of 21 ft. 2 in. aft and 23 ft. forward were recorded.

Her penultimate move towards complete loading brought her 1,115 tons of wheat in No. 5 hold and 925 tons in No. 6, No. 82, and her draughts were 24 ft. 3 in. aft and 22 ft. 5 in. forward.

For final draught and trim, 560 tons of wheat were added to No. 2 hold, No. 83, giving departure draughts of 23 ft. 11 in. aft, 23 ft. 9 in. amidships and 23 ft. 7 in. forward which indicated no sag or hog on a trim aft of 4 in. At the date of this loading the maximum permissible draught for the Sault Ste. Marie channels was 23 ft. 10 in. and on arrival at the Sault Lock the ship had lost 1 in. aft while showing $\frac{1}{2}$ in. of sag.

Some of the actual recorded draughts show inconsistencies when displacement increases are related to T.P.I. and the explanation given was that some of the readings must have been taken when the ship was partly aground which is a common feature in these operations. Nos. 79, 80 and 81 were obviously aground forward and with these exceptions the calculated draughts and trims correspond reasonably well with the actual. It is worthy of note that good propeller immersion was maintained throughout the entire operation.

The foregoing history of one ship's experience does not portray all the unusual and detrimental conditions possible

TABLE 7

GRAIN CARGO DISTRIBUTION, GROUP 4

LOADED AT PORT ARTHUR

Condition	CARGO HOLDS AND BALLAST TANKS												TOTALS (Tons)		
	6		5		4		3		2		1		Cargo	Ballast	Ballast and Cargo
	Cargo	Ballast	Cargo	Ballast	Cargo	Ballast	Cargo	Ballast	Cargo	Ballast	Cargo	Ballast			
15	—	1900	—	1900	—	1400	—	1900	—	1900	—	1852	—	10852	10852
75	—	1900	—	1900	2000	1400	—	1900	—	1900	—	1852	2000	10852	12852
76	2680	—	2680	—	2000	—	—	1900	—	1900	—	1852	7360	5652	13012
77	2680	—	2680	—	2000	—	—	1900	840	1900	—	1852	8200	5652	13852
78	2680	—	2680	—	2000	—	—	1900	840	1900	1000	1852	9200	5652	14852
79	2680	—	2680	—	2330	—	3540	—	840	1900	1000	1852	13070	3752	16822
80	2680	—	2680	—	2330	—	3540	—	840	1900	3805	—	15875	1900	17775
81	2680	—	2680	—	2330	—	4395	—	3840	—	3805	—	19730	—	19730
82	3605	—	3795	—	2330	—	4395	—	3840	—	3805	—	21770	—	21770
83	3605	—	3795	—	2330	—	4395	—	3840	—	3805	—	22330	—	22330

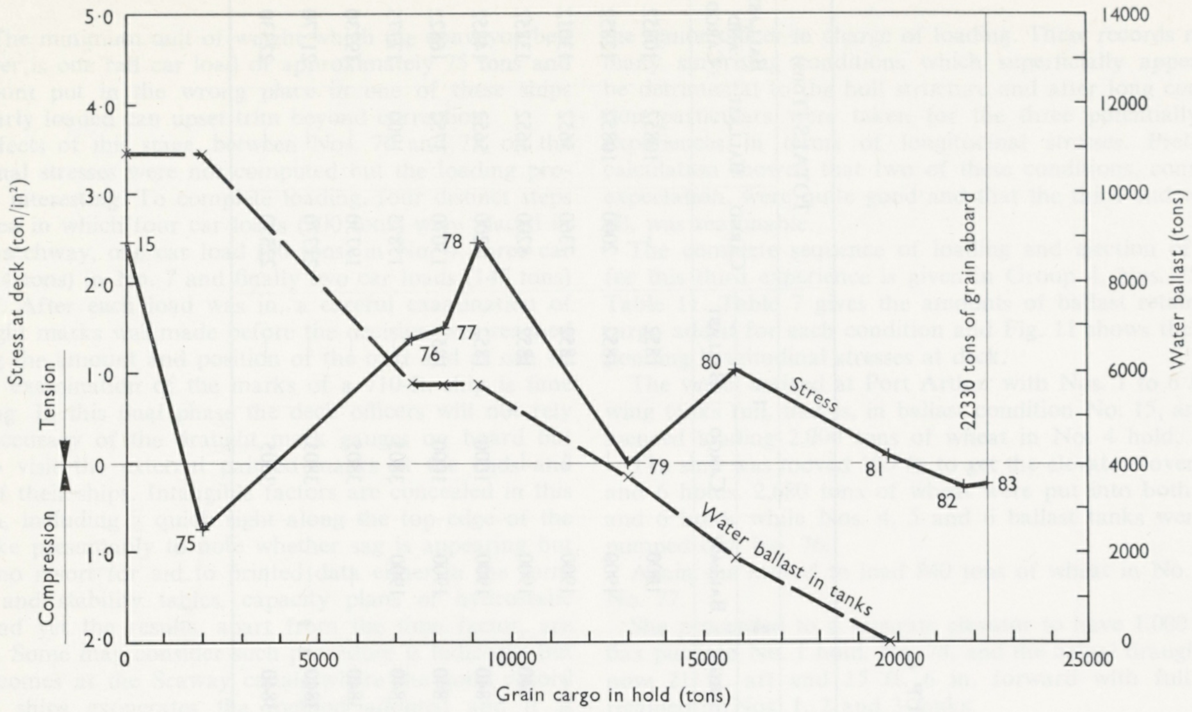


FIG. 11

Load stresses with grain at Port Arthur

but the general pattern of working ballast while loading is amply illustrated and this accepted practice is controlled principally on requirements for draught and trim and the avoidance of premature sag. Under these conditions which are all taking place in still water, it is difficult to imagine that the operators could, unknowingly, overstress their ships. The large lakers with 1,200,000 cubic feet of cargo space find difficulty in obtaining complete cargoes at one elevator from which arises the need to move around and suffer piecemeal loading. In many cases the nature of the cargo and sequence of loading are unknown and unpredictable in advance and the inherent and unerring ability of the ships' officers to manipulate ballast in relation to cargo intake, without the complication of loading diagrams is a vital factor in the successful operation of these large ships.

It is interesting to note that, in Fig. 11, the first 2,000 tons placed in No. 4 hold, No. 75, produced a small sag condition which is contrary to the result desired by the operators but succeeding loads eliminate this. The stresses induced during loading are each moderate and the departure condition, No. 83, is ideal for operation in the open lakes and for canalling.

Loading ore at the head of the lakes takes a variety of forms. New ore docks on Lake Superior have conveyor belt loaders and draughts are virtually unrestricted so that the procedure described for Seven Island is applicable. Ore is mainly in pellet form at these new docks.

At older ore docks with fixed chutes where heights are restricted, ballast might be retained until the first run of ore is completed or about half the total cargo is aboard, before discharge, even to the extent of interrupting loading to complete ballast discharge. The conditions produced by this type of operation could, in theory at least, produce high bending moments but again the overall requirements of draughts and trim are such that they work in favour of reasonable longitudinal stresses.

When unloading ore or grain, ballast and cargo are worked together and a myriad of conditions is possible. In general, ballast is added to control draught and trim as cargo is removed and the entire operation is in still water. Ore is removed more or less uniformly along the length of the ship and this, in association with the ballast added, never appears to excessively upset the rough balance of weights against buoyancy. Unloading grain, however, can be more severe on the structure. For example, it is possible that in No. 74, No. 2 hold containing wheat could be completely emptied before commencing to discharge the remaining cargo which is entirely barley. In this event No. 2 ballast tank would be filled which immediately eases the situation and depending on draught and trim, Nos. 1 or 3 ballast tanks might be partially filled. Longitudinal bending moments appear to be linked indivertibly to draught and the constant desire to maintain draught and trim to within the range of accepted practice automatically works against the production of high longitudinal stresses.

This is not to say that high stresses will never be produced but if they are, it will be obvious and known to the operators that the accepted code of practice is not being applied and that the facilities provided on board ship for eliminating or minimising the condition are not being customarily employed.

Draught looms large in the life of the large lake ship and further explanation of the influence of this premier factor on sag and longitudinal stresses may be opportune.

With deadweight cargoes such as ore, when the whole volume of cargo space is not required, sag is nearly always produced in the loaded conditions. This sag could, theoretically, be increased to produce a tensile stress in the bottom plating of an amount which might damage the structure but, in practice, because limiting draughts are reached when loaded, sag must be minimised for transit in canals and their approaches and in restricted rivers. The parallel middle body

is long in these ships, there is only a 3-in. rise of floor, the bilge radius is small and the beam is 75 ft. At maximum draught with sag the bilges are liable to ground in negotiating some of the confined channels especially when passing other large vessels. In addition, the ships squat in these shallow passages further reducing the small clearance between the bilge and a possible rock bottom. Thus sag, which, although profitable in terms of cargo carried, is treated with great respect by the masters of these ships and is consequently limited to comparatively small amounts.

Grain varies considerably in weight per bushel and often the limiting draught cannot be reached when the holds and hatchways are full. In such a case an attempt is made to trim the vessel so that the after draught is on the limiting draught while producing a slight hog. This procedure in loading gives better speed, minimises grounding of bilges and gives better manœuvrability with greater margin against fore end squat in restricted channels. Condition No. 74, in which, by virtue of the light weight barley cargo, the limiting Sault draught was not obtained, is an excellent example of this type of loading and emphasises that hog is always preferred to sag. Also, condition No. 83, which came much closer to the limiting Sault draught, departed from Port Arthur with no sag.

Miscellaneous Conditions

Group 5 *Condition Nos.* 84 *to* 87, *Table* 12

No. 84 is the completely dry condition with no consumable items, no water ballast and no cargo and only exists between launching and handing over. The remaining three conditions could also apply before handing over but before maximum fuel and water are aboard, No. 87, ballast would be present and hogging stresses reduced. These conditions might also be nearly obtained when dry docking although ballast would be retained as long as possible.

A condition between Nos. 84 and 85 might arise during winter lay up when all water is removed but to winter in any of these conditions would be a most grievous state of affairs for the owners who can usually obtain winter storage grain for their large ships and a condition resembling No. 74 would apply.

A few ships have been laid up for short periods during the summer due to lack of cargoes and when this misfortune befalls, all ballast is normally discharged and they lay up in a condition between 85 and 86.

Conditions between Nos. 85 and 86 apply to post-war ore-loading berths where loading gear is high enough to avoid obstruction and time permits of the discharge of all ballast whilst at the quay wall. No. 62, Table, 11, is such a case and the manner in which the ship came to be in this condition is described in Group 6.

A recent development has been the fitting of bow thruster manœuvring gear in the double bottom immediately aft of the fore peak bulkhead. External examination of this equipment may at some time be necessary and the draught forward in No. 85 is small enough to permit access to the tube.

In all of the conditions in Group 5 the ship is practically stationary in still water and the maximum stress of 5·87 tons per sq. in. at deck which represents the outside fringe of practical operation is well within the total permissible stress which could apply in these cases.

Large lakers would never venture into the open lakes or through a canal on any of the draughts shown in this Group. They covet propeller immersion and can satisfy their desire so easily that the point has been reached where it would appear to be an irrevocable rite of modesty to keep one's propeller covered in public. Canalmen are amazed, and their comments are not suitable for reproduction, at some sea-going ships passing through the canals in ballast with more than half a blade showing and it is obvious, even to the casual observer, that these "salties" are not sufficiently under control for efficient manœuvrability in windy weather by the manner in which they yaw when approaching, and when within, the locks. The large laker on the other hand always appears to be composed and under control even in adverse conditions.

Group 6, *Condition Nos.* 88 *to* 110, *Table* 12

This group may be termed transition conditions in which the laker with no ballast and no cargo, Nos. 85, 86 or 87, would take on ballast to achieve operational draughts. The quantity of ballast in each condition is given in Table 8 and the resultant stresses at the deck, in the form of curves on a base of water ballast in tanks, are shown in Fig. 8.

The procedure for ballasting from zero is fairly uniform throughout the large laker fleets and, without complication, is based on getting the required amount of water into the tanks in the shortest possible time. Broadly this gives two extremes within which all normal transition conditions would fall and these boundary cases are examined in detail.

Before commencing to fill, the draughts necessary for the trip on hand are mentally determined and an approximate estimate of the amount of ballast required to give these desired draughts is culled from experience. It may be the order of these two estimates should be reversed or possibly they form a single thought in the Master's mind but whatever the process the decision requires no visible effort.

Let it be assumed the ship is in condition No. 86 and it is required to obtain a large draught forward which will include hold flooding with Nos. 1 to 6 ballast tanks full. Then as a preliminary all six filling valves (port and starboard) would be opened wide and the ballast tanks allowed to fill together. Because of the distances of the outlets from the manifold, No. 6 tank fills faster than 5, 5 faster than 4 and so on. The rate of fall off varies from ship to ship and an approximate 5 per cent difference between adjacent tanks was assumed for the subject vessel. If filling is allowed to continue until the six ballast tanks are full Nos. 88, 89, 90 and 91, Fig. 8, would each be obtained in turn and finally No. 15 would be reached. Hold flooding would normally have commenced earlier than No. 15 and as was seen in Group 2 hold flooding reduces hogging stresses after the first few feet is in the aft end of the hold. This sequence of conditions represents the upper limit of the two extremes mentioned above.

The lower limit is illustrated in Fig. 8, by Nos. 86 and 92 to 97 in which Nos. 3, 4, 5 and 6 ballast tanks are filled together and if continued until these four tanks are full condition No. 4 would be reached.

Until about the first 5,000 tons of water ballast is in the tanks all the transition conditions fall between or close to these two extremes. After this point is reached and depending on the required draughts, No. 6 tank may be stopped or delayed in filling and similarly the forward tank of any group filling together might also be stopped but the intermediate tanks of the group would normally be allowed to fill completely. This procedure was shown for the ballast conditions in Group 1 where hold flooding is not included and in which the after draught is controlled by the forward or after tanks.

TABLE 8

BALLAST DISTRIBUTION, GROUP 6
TRANSITION CONDITIONS

Condition	Ballast Tanks						Fore Peaks		Total Ballast (Tons)
	6	5	4	3	2	1	2	1	
88	562	538	512	488	462	438	—	—	3000
89	1125	1075	1025	975	925	875	—	—	6000
90	1406	1344	1281	1219	1156	1094	—	—	7500
91	1718	1641	1400	1490	1414	1337	—	—	9000
92	268	256	244	232	—	—	—	—	1000
93	536	512	488	464	—	—	—	—	2000
94	804	768	732	696	—	—	—	—	3000
95	1072	1024	976	928	—	—	—	—	4000
96	1340	1280	1220	1160	—	—	—	—	5000
97	1632	1558	1400	1410	—	—	—	—	6000
98	—	—	700	—	—	—	—	—	700
99	—	—	1400	—	—	—	—	—	1400
100	—	974	1400	926	—	—	—	—	3300
101	675	1618	1400	1507	—	—	—	—	5200
102	—	1900	1400	1900	—	—	—	—	5200
103	950	1900	1400	1900	—	—	—	—	6150
104	950	—	—	—	—	—	—	—	950
105	1900	—	—	—	—	—	—	—	1900
106	1900	950	—	—	—	—	—	—	2850
107	1900	1900	—	—	—	—	—	—	3800
108	1900	1900	700	—	—	—	—	—	4500
109	1900	1900	1400	—	—	—	—	—	5200
110	1900	1900	1400	950	—	—	—	—	6150

Incidentally, the small gradual increase to the forward draught should be noted in Nos. 92 to 97, Table 12, showing that this sequence of ballasting increases the after draught in a hurry and makes it favourite in fine weather when a light forward draught can be used. While the stresses induced by all normal transition conditions would fall within the angle formed by these two extremes, the position of the apex at No. 86 is, of course, variable in magnitude depending on the oil fuel, fresh water, etc., carried and as previously explained would seldom reach the value of 5·87 tons per sq. in. obtained when the maximum capacity of consumable items is on board.

Two other courses of filling are included to illustrate the best and worst modes of ballasting, neither of which is likely to be adopted because they do not comply with the maxim of getting the water into the ship in the shortest possible time.

Nos. 86, 98 to 101 and 4 show the best route for reducing the hogging stresses and is accomplished by first filling No. 4 tank completely, then Nos. 3 and 5 tanks together until each is about half full and coming to condition No. 4 by filling Nos. 3, 5 and 6 tanks together.

The size of the filling lines in relation to the capacity of the two pumps is such that two tanks can roughly be filled in the same time as one and no large laker would be content to commence filling on one tank only whatever desirable reward might accrue. Although there are better ways of ballasting, so far as longitudinal stresses are concerned, the practice adopted is sound and efficient and there is consequently no real reason for attempting to change it.

No. 102 shows the greatest sagging stress which could be induced in ballast when Nos. 3, 4 and 5 tanks are full and Nos. 6, 2 and 1 tanks are empty. No. 103 is the same condition with No. 6 tank half full.

Nos. 86, 104 to 110 and 4 show the course the stress at deck takes when the ballast tanks are filled separately and in sequence, commencing with No. 6 tank. No. 6 tank completely full with the remainder empty, No. 105, produces the highest stress obtainable and this is not excessive in still water. But again, filling tanks singly is such a time consuming operation that the purists of naval architecture who consider all things possible can rest assured that no Master on the lakes could be talked into even considering this sequence of ballasting.

The transition conditions in Group 6 have so far been considered on the common precept of filling but, of course, these conditions hold for the reverse operation of discharging all ballast from any of the conditions in Group 1. The basic principle applying in these cases is to complete discharge in the shortest possible time which simply means that all the tanks containing water will be made to empty together. There is one difference in this operation, however, in that No. 6 tank empties quicker than No. 5, No. 5 quicker than No. 4, etc., and so the distribution of ballast for conditions Nos. 88 to 97 is not quite the same when emptying as that tabulated in Table 8 for filling. This redistribution widens the angle between the two extreme sequences, Fig. 8, increasing the stresses at the upper limit and reducing the stresses at the lower limit but the differences are not great.

No. 62, Table 11, was the light ship condition of a ship alongside the quay wall at Seven Islands with loading imminent and the ballasting procedure preceding this condition portrays the general practice. This ship passed through the St. Lawrence Section of the Seaway in fair weather with Nos. 1 to 6 ballast tanks full, fore peaks empty and 4 ft. of water on her after hold bulkhead. Except for the quantity of fuel on board this condition is identical to No. 39, Table 10, and the actual draughts, obtained at Iroquois Lock, were 20 ft. 11 in. aft, 16 ft. midships and 11 ft. 11 in. forward which indicate 5 in. hog on 9 ft. trim. This canalling condition was maintained until clear of St. Lambert Lock and well down river. With weather remaining fine and the river widening ballast was discharged to reduce the forward draught and this operation resulted in the hold and No. 1 tank being emptied and only 1,510 tons of ballast being retained in No. 6 tank with Nos. 2 to 5 tanks full. This is almost No. 9, Tables 10, and 4, and is the condition in which she sailed into Seven Islands Harbour.

When the large laker arrives at a loading port and a number of ships are ahead, she will anchor, when possible, and complete discharge of all hold ballast, if any remains, but she will retain sufficient water ballast in her tanks to maintain good propeller immersion with sufficient draught forward for manœuvring into position. When her time comes she will proceed to a waiting quay directly behind the ship being loaded and on being secured to the quay wall, not before, she will commence discharge of all her double bottom and wing tanks. If time permits, and it usually does, she will be completely dry of water ballast, as shown in No. 62, when towed forward on her winches under the loading equipment.

Group 7 *Condition Nos.* 111 *to* 124, *Table* 12

Mandatory dry docking for lakers is once in four years and when necessary interim examination of propeller and stern bearing might be carried out by tipping the vessel afloat. A draught aft of about 8 ft. would normally lift the propeller boss clear and less than 7 ft. provides ample working space.

The first of the over 700-ft. lakers is four years old this year and the Author is not aware of any which have been tipped for propeller examination although it is probable this has been done in the upper lakes. The practice is common enough with smaller lakers and the conditions investigated in this Group may be useful for the large vessels.

With minimum oil fuel, fresh water, etc., on board it was found that the quickest and simplest method of tipping was to fill Nos. 1 and 2 ballast tanks, No. 111, Table 12, which gives a sufficiently reduced draught aft and low enough hogging stress at deck to conclude that the operation would be successful. No. 112 shows the reduction in the after draught and corresponding increase in stress due to filling No. 2 fore peak tank.

The ballast distributions for this group are given in Table 9 and the stresses at deck in Fig. 12.

Various alternative arrangements of ballast using the two fore peaks and Nos. 1, 2 and 3 ballast tanks were examined, Nos. 113 to 120, but none of these gives such a good combination of draught aft and stress at deck as No. 111 although the stresses obtained are not excessive for the vessel moored in still water. Possibly the greatest attraction to using No. 111 is neither of the slow filling peak tanks need be used so that ballasting can be done entirely on the main ballast lines. A few hundred tons of hold ballast could be used in preference to the fore peak tanks but since this would wedge in No. 1 hold some caution would require to be exercised as its removal would not be possible during repairs in the area of the tail shaft.

A ship arriving in ballast with an operational trim aft and intending to change trim for tipping would transfer water to the forward tanks and pump out excess ballast simultaneously. The transition from any of the ballast conditions in Group 1

TABLE 9

BALLAST DISTRIBUTION, GROUP 7
TIPPING FOR PROPELLER EXAMINATION

Condition	Ballast Tanks 6	5	4	3	2	1	Fore Peaks 2	1	Total Ballast (Tons)
111	—	—	—	—	1900	1852	—	—	3752
112	—	—	—	—	1900	1852	650	—	4402
113	—	—	—	—	—	—	650	210	860
114	—	—	—	—	—	926	650	210	1786
115	—	—	—	—	—	1852	650	210	2712
116	—	—	—	—	950	1852	650	210	3662
117	—	—	—	—	1425	1852	650	210	4137
118	—	—	—	—	1900	1852	650	210	4612
119	—	—	—	475	1900	1852	650	210	5087
120	—	—	—	950	1900	1852	650	210	5562
121	—	—	—	—	1900	1852	—	—	3752
122	—	—	—	—	1900	1852	650	—	4402
123	—	—	—	—	1900	1852	650	210	4612
124	—	—	—	—	950	1852	650	210	3662

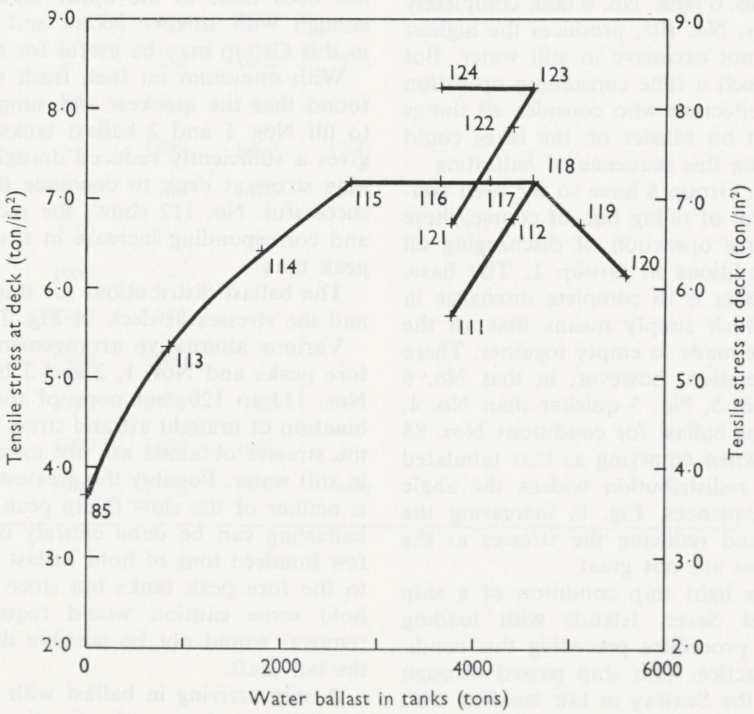


FIG. 12

Stresses due to tipping for propeller examination

to the finally tipped condition would be such that the longitudinal stresses would increase gradually until the final trim by the head is established. As the suctions are always placed at the aft end of the ballast tanks a wedge of water might be retained in each pumped out tank when trimming by the head and the amount depends on trim. This water was not included in estimating the weight bending moments in this Group and from the positions of these wedges they tend to reduce the hogging stresses while increasing the draughts aft given in Table 12.

Nos. 121 to 124 were taken with mean or "departure" capacities for oil fuel, fresh water, etc. These conditions show that to obtain a sufficiently reduced draught aft, either or both the fore peak tanks would require to be filled in association with Nos. 1 and 2 ballast tanks and that the longitudinal stresses are becoming high. These stresses are within the theoretical permissible maximum for a ship tied to a quay wall in still water but the conditions also show that the tipping operation should be carried out with the least possible amount of consumable items in the after part of the ship.

Deviations

In all conditions of ballast the vessel hogs yet this method of analysis presupposes that the hull girder remains straight all fore and aft. When a ship hogs the mean moment of weight is in excess of the mean buoyancy moment but when hogging the buoyancy moment is greater than when straight and so in each ballast case investigated it is reasonable to conclude that the still water bending moments obtained are over estimated in proportion to the amount of hog.

The position of the oil fuel bunkers and boilers affects stresses to a small degree. In the subject vessel it was assumed that the bunkers and boilers are aft of the main engines, Fig. 6, but in some ships these are in a forward position in the engine room. The aft position assumed tends to reduce sag when loaded and increases ballast hog while the forward position, although acting similarly, does so with reduced magnitude. When loading ore sag can be eliminated without difficulty by planned disposition of the cargo but grain is less accommodating and may be placed on board in positions dictated by factors other than proper loading with respect to sag. About 40 per cent of the mileage covered by these ships is in ballast so that all phases of ballasting are of high importance and in all of the conditions in which the ships can successfully operate hog is produced. From the viewpoint of longitudinal stresses, therefore, the forward position is to be preferred although where grain is the cargo and the mandatory draught is obtained, a slight benefit in deadweight may sometimes be obtained if the bunkers and boilers are positioned aft.

Instances of loading and ballasting occur outside the general code of practice but how far these conditions prejudice the safety of the structure is another matter since enquiry regarding some of these has failed to obtain sufficient data to complete a reliable investigation into the longitudinal stresses induced. Of known cases, some are vague and clouded in hearsay while others are more definite and the following example, details of which are nearly complete, might give some idea of this indeterminate problem.

At the close of navigation a large laker was moored to a quay wall under an elevator at Prescott with a full cargo of storage grain, the intention being that she would discharge the cargo in the spring. She was, therefore, fully winterised and had a slight trim aft. During winter grain is moved by rail and it was found that the shore storage bins, which at commencement of winter were completely full, exhausted their supply of a certain type and grade of grain which this ship had in No. 2 hold and since she was conveniently under an elevator it was decided to empty this hold. The matter was urgent, the temperature was sub-zero and the ship was embedded in thick ice. Normally water ballast would be used to partly compensate for cargo removed but with low temperatures, in addition to machinery being closed down, this was impossible and so No. 2 hold was emptied without adding ballast and without first freeing the ship from the surrounding ice. Towards the end of this operation with No. 2 hold nearly empty the ship, which had so far remained perfectly motionless, suddenly broke through the ice and in a surprisingly short time reduced her mean draught by about 3 ft. and changed trim by about 11 ft., all of which was accompanied by a tremendous and frightening cacophony. She remained in this condition until spring when on discharge of the remaining cargo holds and clearing of the snow and ice from her deck, examination revealed no evidence of damage. Treatment such as this tends to make academic estimates of structural requirements somewhat futile.

CONCLUSIONS

Two opposite conditions concern longitudinal strength, one being sag and the other hog.

Sagging conditions exist only when fully loaded. In the large ships load draught is controlled by canals and restricted waterways so that draughts aft and forward are equal and sag must be limited to give a midship draught not noticeably different from the ends otherwise the ship might be refused entry to the Seaway. In addition, sag when loaded increases the probability of grounding damage in restricted channels. It can be said, therefore, with good reason that while, theoretically, excessive sagging moments could be produced in service by haphazard or even uniform loading, the controlling factors in the actual operation all tend to minimise these moments and in practice compulsorily judicious loading so reduces them that they become of secondary importance in structural design.

Where, because of light density cargoes, the maximum permitted draught cannot be obtained when fully loaded the tendency is to induce hog rather than sag so that in no condition does the longitudinal stress due to sag loom large as a potential danger to the structure.

In all conditions of ballast the vessel hogs but the stresses produced in operational voyages in the open lakes are moderate and sufficiently low to take the superincumbent stresses due to storm waves without exceeding the safe total.

The ballast conditions which produce high stresses are those when the ship is in sheltered or protected localities such as loading or unloading berths where there is no danger of wave stresses also being experienced and again the total stress is safe.

The nature and magnitude of these hogging conditions with the associated tensile stresses at the deck emphasise that the design and workmanship of the topsides in general and the hatchway corners and sides in particular, are of extreme importance. It is imperative that discontinuities and notches be avoided.

The present arrangement of longitudinal stiffening on the bottom is suited to the high compressive stresses produced by hogging and cannot be effectively improved in principle.

TABLE 10

OPERATIONAL BALLAST CONDITIONS

Condition		Summary of Weights: Light Ship	Fuel, etc.	Ballast in tanks	Ballast in Holds	Displacement	Calculated L.C.G. from)(	Draughts: Aft	Amidship	Fwd.	Trim / L	Draught / L	Values of "a"	Mean Moments: Buoyancy	Weights	Still Water Bending Moments	Stress Tons/in.²: Deck	Keel
GROUP 1	1			5630	—	13675	51·9a	20– 5	11– 8	2–11	·02465	·01644	·20764	1008014	1040518	32504h	0·77t	0·65c
	2			6430	—	14475	60·2a	22– 6	12– 4	2– 2	·02864	·01737	·20805	1069091	1120788	51967h	1·24t	1·03c
	3			6230	—	14275	45·7a	20– 4½	12– 1	3– 9½	·02236	·01701	·20847	1056448	1055205	1243s	0·03c	0·02t
	4			7100	—	15145	54·6a	22– 7½	12– 9½	2–11½	·00270	·01801	·20893	1123307	1142466	19159h	0·46t	0·38c
	5			7400	—	15445	34·8a	20– 3½	12–11	5– 6½	·02101	·01820	·21006	1151754	1157023	5269h	0·13t	0·10c
	6			8050	—	16095	41·5a	22– 0	13– 5½	4–11	·02406	·01896	·21047	1202568	1222218	19650h	0·47t	0·39c
	7			8138	—	16183	29·2a	20– 4½	13– 5½	6– 6½	·01948	·01896	·21102	1212303	1243939	31636h	0·75t	0·63c
	8			8513	—	16558	33·1a	21– 4	13– 9	6– 2	·02136	·01937	·21122	1241570	1281552	39982h	0·95t	0·79c
	9			8575	—	16620	25·4a	20– 4	13– 9½	7– 3	·01843	·01942	·21160	1248461	1259343	10882h	0·26t	0·22c
	10			9000	—	17045	29·8a	21– 5	14– 1½	6– 7	·02054	·01990	·21185	1281899	1301971	20072h	0·48t	0·40c
	11			9313	—	17358	20·7a	20– 4½	14– 3½	8– 2½	·01714	·02013	·21237	1308643	1346259	37616h	0·89t	0·75c
	12			9463	—	17508	22·2a	20– 9	14– 5	8– 1	·01784	·02037	·21247	1320573	1361304	40731h	0·97t	0·81c
	13			9926	—	17971	15·0a	20– 1	14– 8½	9– 4	·01514	·02072	·21296	1358622	1420638	62016h	1·47t	1·23c
	14			10389	—	18434	8·2a	19– 5½	15– 0	10– 6½	·01256	·02113	·21345	1396812	1479971	83159h	1·98t	1·65c
	15			10852	—	18897	1·7a	18– 9½	15– 3½	11– 9½	·00986	·02154	·21394	1435203	1539305	104102h	2·47t	2·07c
	16			11502	—	19547	9·1f	17– 7	15– 9	13–11	·00516	·02218	·21469	1489774	1644410	154636h	3·68t	3·07c
	17			11712	—	19757	12·6f	17– 2	15–10½	14– 7	·00364	·02235	·21490	1507252	1680425	173173h	4·12t	3·44c
	18			8513	146	16704	34·7a	21– 8½	13–10½	6– 0½	·02207	·01954	·21131	1253051	1297750	44699h	1·06t	0·89c
	19			8513	542	17100	38·3a	22– 8½	14– 2½	5– 8½	·02394	·02013	·21170	1285124	1334749	49625h	1·18t	0·99c
	20			9000	151	17196	31·5a	21– 9½	14– 2½	6– 7½	·02136	·02001	·21187	1293377	1318641	25264h	0·60t	0·50c
	21			9000	561	17606	35·0a	22– 9	14– 6	6– 3	·02324	·02042	·21198	1324903	1356472	31569h	0·75t	0·63c
	22			9000	1156	18201	38·9a	24– 0½	15– 0½	6– 0½	·02535	·02118	·21235	1372069	1402023	29954h	0·71t	0·59c
	23			9000	1902	18947	42·3a	25– 5	15– 7½	5–10	·02758	·02201	·21280	1431332	1449508	18176h	0·43t	0·36c
	24			9463	173	17681	24·1a	21– 2½	14– 6½	7–10½	·01878	·02048	·21245	1333497	1380040	46543h	1·11t	0·92c
	25			9463	632	18140	28·0a	22– 3	14–11	7– 7	·02066	·02107	·21269	1369660	1420428	50768h	1·21t	1·01c
	26			9463	1289	18792	31·9a	23– 7	15– 5½	7– 4	·02289	·02177	·21300	1420957	1466762	45805h	1·09t	0·91c
	27			9463	2091	19599	35·3a	24–11½	16– 0½	7– 1½	·02512	·02259	·21343	1484970	1514923	29953h	0·17t	0·59c

GROUP 2

7450 TONS IN ALL CONDITIONS

595 TONS IN ALL CONDITIONS

28	9926	201	18172	17·2a	20– 7½	14–10½	9– 1½	·01620	·02094	·21300	1374076	1441813	67737h	1·61t	1·34c
29	9926	716	18687	21·3a	21– 8½	15– 3	8– 9½	·01819	·02148	·21319	1414278	1484505	70277h	1·67t	1·39c
30	9926	1433	19402	25·3a	23– 2	15–10½	8– 7	·02054	·02235	·21360	1471215	1531311	60096h	1·43t	1·19c
31	9926	2336	20307	28·5a	24– 7	16– 6½	8– 6	·02265	·02330	·21413	1543660	1582353	38693h	0·92t	0·77c
32	9926	3381	21352	30·5a	26– 0	17– 4	8– 8	·02441	·02441	·21484	1628479	1640771	12292h	0·29t	0·24c
33	10389	237	18671	10·7a	20– 1	15– 2½	10– 4	·01373	·02154	·21361	1415851	1504039	88188h	2·10t	1·75c
34	10389	822	19256	15·0a	21– 3½	15– 8	10– 0½	·01585	·02207	·21378	1461374	1548805	87431h	2·08t	1·74c
35	10389	1644	20078	19·0a	22– 9	16– 4	9–11	·01808	·02300	·21423	1526965	1597584	70619h	1·68t	1·40c
36	10389	2657	21091	21·7a	24– 2	17– 1	10– 0	·01995	·02406	·21485	1608647	1653957	45310h	1·08t	0·90c
37	10389	3906	22340	22·7a	25– 6½	18– 0½	10– 6½	·02113	·02541	·21574	1710969	1726522	15553h	0·37t	0·31c
38	10852	290	19187	4·6a	19– 6½	15– 7	11– 7½	·01115	·02194	·21407	1458113	1567174	109061h	2·59t	2·71c
39	10852	957	19854	9·1a	20–10½	16– 1	11– 3½	·01350	·02265	·21433	1510634	1613357	102723h	2·44t	2·04c
40	10852	1872	20769	12·8a	22– 4½	16–10	11– 3½	·01561	·02370	·21487	1584235	1665097	80862h	1·92t	1·61c
41	10852	3099	21996	14·6a	23– 9½	17– 9	11– 8½	·01702	·02500	·21565	1683920	1735343	51423h	1·22t	1·02c
42	10852	3764	22661	15·1a	24– 5½	18– 3	12– 0½	·01749	·02570	·21610	1738450	1777000	38550h	0·92t	0·77c
43	10852	4284	23181	14·2a	24– 8½	18– 7	12– 5½	·01725	·02617	·21642	1780975	1811863	30888h	0·73t	0·61c
44	10852	5032	23929	11·8a	24–11	19– 1½	13– 4	·01631	·02694	·21697	1843116	1867257	24141h	0·57t	0·48c
45	10852	5723	24620	9·7a	25– 1½	19– 8	14– 2½	·01538	·02770	·21750	1900972	1918471	17499h	0·42t	0·35c
46	10852	6443	25340	7·7a	25– 3½	20– 2	15– 0½	·01444	·02840	·21797	1960793	1972137	11344h	0·27t	0·23c
47	10852	7204	26101	5·6a	25– 6	20– 8½	15–11	·01350	·02917	·21847	2024311	2028863	4552h	0·11t	0·09c
48	11502	461	20008	5·2f	18– 7½	16– 1½	13– 7½	·00704	·02271	·21484	1525974	1680709	154735h	3·68t	3·07c
49	11502	1454	21001	1·3f	20– 1½	16–11	13– 8½	·00904	·02383	·21536	1605585	1737356	131771h	3·13t	2·62c
50	11502	2219	21766	1·0f	20– 9	17– 5½	14– 2	·00927	·02459	·21579	1667394	1785338	117944h	2·80t	2·34c
51	11502	2864	22411	2·6f	20–11½	17–11	14–10½	·00857	·02523	·21621	1720146	1832756	112610h	2·68t	2·24c
52	11502	3517	23064	4·1f	21– 2½	18– 5	15– 7½	·00786	·02594	·21665	1773870	1881033	107163h	2·55t	2·13c
53	11502	4172	23719	5·6f	21– 5½	18–11	16– 4½	·00716	·02664	·21708	1827867	1929488	101621h	2·42t	2·02c
54	11502	4827	24374	7·0f	21– 8½	19– 5	17– 1½	·00646	·02735	·21750	1881977	1978137	96160h	2·29t	1·91c
55	11502	5487	25034	8·4f	21–10	19–10	17–10	·00563	·02793	·21785	1936048	2026981	90933h	2·16t	1·81c
56	11502	6181	25728	9·8f	22– 1	20– 4½	18– 8	·00481	·02870	·21829	1993739	2078553	84814h	2·02t	1·68c
57	11712	584	20342	8·5f	18– 4	16– 4½	14– 5	·00552	·02307	·21515	1553686	1720817	168131h	3·97t	3·32c
58	11712	1170	20927	6·8f	19– 0½	16– 9½	14– 6½	·00634	·02365	·21542	1600373	1754378	154005h	3·66t	3·06c
59	11712	1971	21728	6·8f	19– 8½	17– 5	15– 1½	·00646	·02453	·21591	1665409	1808575	143166h	3·40t	2·84c
60	11712	3134	22891	9·1f	20– 2	18– 2½	16– 3	·00552	·02565	·21659	1760076	1894025	133949h	3·18t	2·66c
61	11712	4761	24518	13·2f	20– 7	19– 6	18– 5	·00305	·02746	·21765	1894402	2016335	121933h	2·90t	2·42c

TABLE 11

LOADED CONDITIONS

Condition		SUMMARY OF WEIGHTS				Displace-ment	Calcu-lated L.C.G. from)(	DRAUGHTS			Trim / L	Draught / L	Values of "a"	MEAN MOMENTS		Still Water Bending Moments	STRESS Tons/in.²	
		Light Ship	Fuel, etc.	Cargo	Ballast in Tanks			Aft	Amidship	Fwd.				Buoyancy	Weights		Deck	Keel
GROUP 3	62	7450 TONS IN ALL CONDITIONS	313	—	—	7763	34·0a	10– 9½	6– 8½	2– 7½	·01150	·00945	·20265	558476	730706	172230h	4·09t	3·42c
	63		313	3000	—	10763	47·8a	15– 9	9– 1½	2– 6	·01866	·01285	·20475	782322	934106	151784h	3·61t	3·01c
	64		313	6000	—	13763	41·8a	18– 10	11– 4½	3–11	·02101	·01602	·20784	1015478	1132107	116629h	2·77t	2·32c
	65		313	9000	—	16763	25·6a	20– 0	13– 6½	7– 1	·01819	·01907	·21129	1257358	1332508	75150h	1·79t	1·49c
	66		313	12000	—	19763	4·9a	19– 7½	15– 7	11– 6½	·01138	·02195	·21406	1501816	1525822	24006h	0·57t	0·48c
	67		313	15000	—	22763	10·4f	19– 4½	17– 8½	16– 0½	·00469	·02494	·21623	1747325	1751046	3721h	0·09t	0·07c
	68		313	18000	—	25763	15·8f	20– 5½	19–10½	19– 3½	·00164	·02799	·21796	1993433	1994645	1212h	0·03t	0·02c
	69		313	21000	—	28763	12·9f	23– 0½	22– 0½	21– 0½	·00282	·03104	·21931	2239345	2254445	15100h	0·36t	0·30c
	70		313	23545	—	31308	15·9f	24– 0½	23–10½	23– 8½	·00047	·03363	·22009	2446155	2380873	65282s	1·55c	1·30t
	71		313	24306	—	32069	15·2f	24– 8	24– 5	24– 2	·00070	·03439	·22032	2508232	2445972	62260s	1·48c	1·24t
	72		226	24306	—	31982	16·0f	24–11½	24–11½	24–11½	—	·03515	·22055	2504039	2432528	71511s	1·70c	1·42t
	73		130	24306	100	31986	16·2f	24–11	24–11½	25– 0	·00012	·03515	·22055	2504352	2428046	76306s	1·81c	1·52t
	74		640	21384	—	29474	14·3f	23– 8	23– 1	22– 6	·00164	·03251	·21975	2299304	2321875	22571h	0·54t	0·45c
GROUP 4	75		595	2000	10852	20897	3·5a	20–10	16–10	12–10	·01127	·02371	·21515	1596076	1562646	33430s	0·79c	0·66t
	76		595	7360	5652	21057	13·5a	22– 9	17– 0½	11– 4	·01608	·02400	·21504	1607475	1665927	58452h	1·39t	1·16c
	77		595	8200	5652	21897	6·5a	22– 3	17– 7½	13– 0	·01303	·02482	·21574	1677041	1736445	59404h	1·41t	1·18c
	78		595	9200	5652	22897	5·0f	20–11	18– 3½	15– 8	·00739	·02576	·21657	1760375	1864895	104520h	2·48t	2·08c
	79		595	13070	3752	24867	8·6f	21– 8	19– 8½	17– 9	·00552	·02776	·21776	1922338	1921883	455s	0·01c	0·01t
	80		595	15875	1900	25820	17·8f	20– 6	20– 4½	20– 3	·00035	·02870	·21835	2001418	2044851	43433h	1·03t	0·86c
	81		595	19730	—	27775	25·2f	20– 2½	21– 9½	23– 4½	·00446	·03069	·21921	2161438	2164898	3460h	0·08t	0·07c
	82		595	21770	—	29815	13·3f	24– 1½	23– 4	22– 6½	·00223	·03286	·21986	2327070	2315813	11257s	0·27c	0·22t
	83		595	22330	—	30375	16·2f	23–10	23– 9	23– 8	·00023	·03345	·22004	2372719	2362825	9894s	0·24c	0·20t

TABLE 12

MISCELLANEOUS CONDITIONS

Condition		Summary of Weights: Light Ship	Summary of Weights: Fuel, etc.	Summary of Weights: Ballast in Tanks	Displacement	Calculated L.C.G. From ⋈	Draughts: Aft	Draughts: Amidship	Draughts: Fwd.	Trim / L	Draugh / L	Values of "a"	Mean Moments: Buoyancy	Mean Moments: Weights	Still Water Bending Moments	Stress Tons/in.²: Deck	Stress Tons/in.²: Keel
GROUP 5	84	7450 TONS IN ALL CONDITIONS	—	—	7450	23·8a	9–10½	6– 6½	3– 2½	·00939	·00921	·20293	536699	682588	145889h	3·47t	2·90c
	85		130	—	7580	27·9a	10– 4½	6– 8½	3– 0½	·01033	·00945	·20302	546307	702749	156442h	3·72t	3·11c
	86		595	—	8045	42·8a	12– 1½	7– 2	2– 2½	·01397	·01010	·20292	579534	775487	195953h	4·66t	3·89c
	87		1179	—	8629	59·2a	14– 2½	7– 8½	1– 2½	·01831	·01086	·20251	619988	866747	246759h	5·87t	4·90c
GROUP 6	88		595 TONS IN ALL CONDITIONS	3000	11045	27·4a	14– 5	9– 5½	4– 6	·01397	·01332	·20639	809250	975967	166717h	3·96t	3·31c
	89			6000	14045	18·4a	16– 8	11– 9	6–10	·01385	·01655	·20959	1045011	1177393	132382h	3·15t	2·63c
	90			7500	15545	15·3a	17– 9	12–10½	8– 0	·01373	·01813	·21099	1164343	1277877	113534h	2·70t	2·25c
	91			9000	17045	12·3a	18– 9½	14– 0	9– 2½	·01350	·01972	·21241	1285288	1387379	102091h	2·43t	2·03c
	92			1000	9045	45·8a	13– 8	7–11½	2– 3	·01608	·01121	·20353	653530	826000	172470h	4·10t	3·42c
	93			2000	10045	48·2a	15– 2½	8– 9	2– 3½	·01819	·01232	·20424	728315	876524	148209h	3·52t	2·94c
	94			3000	11045	50·2a	16– 9	9– 7	2– 5	·0219	·01350	·20516	804427	927037	122610h	2·91t	2·43c
	95			4000	12045	51·8a	18– 3	10– 4½	2– 6	·02218	·01461	·20608	881193	977560	96367h	2·29t	1·91c
	96			5000	13045	53·2a	19– 8	11– 1½	2– 7	·02406	·01567	·20691	958195	1028075	69880h	1·66t	1·39c
	97			6000	14045	54·8a	21– 3	11–11½	2– 8	·02617	·01684	·20783	1036235	1081977	45742h	1·09t	0·91c
	98			700	8745	40·9a	12–10½	7– 8½	2– 6½	·01455	·01086	·20357	631978	783649	151671h	3·60t	3·01c
	99			1400	9445	39·4a	13– 7½	8– 3	2–10½	·01514	·01162	·20424	684812	791821	107009h	2·54t	2·12c
	100			3300	11345	36·4a	15– 9½	9–10	3–10½	·01678	·01385	·20634	831029	872325	41296h	0·98t	0·82c
	101			5200	13245	43·6a	18–10½	11– 3½	3– 8½	·02136	·01590	·20765	976365	992244	15879h	0·38t	0·32c
	102			5200	13245	33·6a	17– 7½	11– 2½	4– 9½	·01808	·01579	·20814	978669	951896	26773s	0·65c	0·53t
	103			6150	14195	44·8a	20– 2½	12– 0½	3–10½	·02300	·01696	·20848	1050578	1047181	3397s	0·08c	0·07t
	104			950	8995	59·5a	14–10½	8– 1	1– 3½	·01913	·01138	·19907	635675	870772	235097h	5·59t	4·67c
	105			1900	9945	72·9a	17– 5	8–10	3	·02418	·01174	·20198	713085	966057	252972h	6·01t	5·02c
	106			2850	10895	75·6a	19– 1½	9– 7	½	·02688	·01350	·20362	787546	1015314	227768h	5·41t	4·52c
	107			3800	11845	77·9a	20–10	10– 4½	Minus 1	·02946	·01461	·20456	860170	1064572	204402h	4·86t	4·06c
	108			4500	12545	74·6a	21– 6½	10–11	3½	·02993	·01538	·20547	915056	1072734	157678h	3·75t	3·13c
	109			5200	13245	71·7a	22– 2½	11– 5½	8½	·03028	·01614	·20632	970111	1080906	110795h	2·63t	2·20c
	110			6150	14195	62·6a	22– 5	12– 1½	1–10	·2899	·01708	·20766	1046445	1111686	65241h	1·55t	1·30c
GROUP 7	111		130	3752	11332	51·4f	6– 5½	9– 2½	11–11½	·00775	·01297	·21342	858559	1099588	241029h	5·73t	4·79c
	112		130	4402	11982	66·1f	5– 1½	9– 7½	14– 1½	·01268	·01356	·21602	918865	1204693	285828h	6·79t	5·68c
	113		130	860	8440	8·4f	8– 5½	7– 2½	5–11½	·00352	·01027	·20618	617757	843869	226112h	5·37t	4·49c
	114		130	1786	9366	32·9f	7– 1½	7–10	8– 6½	·00200	·01103	·20795	691491	962536	271045h	6·44t	5·38c
	115		130	2712	10292	53·0f	5– 9	8– 5	11– 1	·00751	·01186	·21291	777901	1081203	303302h	7·21t	6·02c
	116		130	3662	11242	62·7f	5– 3	9– 1½	13– 0	·01092	·01286	·21506	858285	1160956	302671h	7·19t	6·01c
	117		130	4137	11717	67·0f	5– 0	9– 5½	13–11	·01256	·01332	·21595	898252	1200833	302581h	7·19t	6·01c
	118		130	4612	12192	70·9f	4– 8½	9– 9	14– 9½	·01420	·01373	·21675	938129	1240708	302579h	7·19t	6·01c
	119		130	5087	12667	70·7f	4–11	10– 1½	15– 4	·01467	·01427	·21963	975488	1256098	280610h	6·67t	5·57c
	120		130	5562	13142	70·5f	5– 1½	10– 6	15–10½	·01514	·01479	·21708	1012767	1271488	258721h	6·15t	5·14c
	121		595	3752	11797	38·1f	8– 2	9– 8	11– 2	·00423	·01362	·21294	891779	1172326	280547h	6·67t	5·57c
	122		595	4402	12447	53·0f	6–10½	10– 1	13– 3½	·00904	·01420	·21441	947410	1277431	330021h	7·84t	6·55c
	123		595	4612	12657	57·8f	6– 5	10– 2½	14– 0	·01068	·01438	·21517	966809	1313446	346637h	8·24t	6·88c
	124		595	3662	11707	48·9f	7– 0	9– 7	12– 2	·00728	·01349	·21336	886721	1233693	346972h	8·24t	6·89c

These long ships are work horses and there are occasions when they might be pushed towards the limit of their endurance but over the whole piece they are handled extremely well and give excellent service. They are comparatively young, however, so far as ages go on the lakes and as they advance in years it is doubtful whether they will be permitted to retain their present youthful vigour.

The strength of steel ships has been ardently studied for over 100 years (Ref. 15) and reliable answers to all the problems are not yet known. The persistent struggle for structural economies keeps pace with increase in knowledge and sometimes zealously steps ahead with unfortunate results. When there is produced a trouble-free ship which is also economically successful in operation the designing naval architect deserves to be complimented, not so much for his ability with figures but for the elusive something extra which must have been embodied in the design. Although many people may not agree, structural design is not yet a complete science and requires this extra essential ingredient for full functional maturity. It is analogous to salt in soup and the foregoing examination of large lakers, so far as it goes, merely confirms, as does also the successful operational record, that during the evolution of this unique type of ship just the right amount of salt was added.

It would be a simple matter to conclude that these ships are too strong for their service on the lakes and pursue this by saving a few tons of steel but such action could herald an era of costly and disappointing service results culminating, after a cycle of modifications, in a return to the position held now. It is not intended by this statement that the end of all advancement in large laker structural design is already here. Weight saving and economical features have been evolved in the last year or so, as some builders might agree, and no doubt this gradual advance will continue but when all things are considered the basic longitudinal strength of these ships is just about right and, for the overall good of the industry, this fundamental datum line should be maintained.

ACKNOWLEDGMENTS

The Author would like to express his thanks for the assistance and willing co-operation received from all sections of the Great Lakes shipping industry and particularly to owners for providing the means of acquiring much of the information at first hand and to shipbuilders for supplying necessary data. To list the names of the many people involved would be long and to mention only a few would do less than justice to the remainder.

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LIST OF TABLES AND ILLUSTRATIONS

APPENDIX

Comparison of still water bending moments induced in a 600 ft. × 65 ft. × 33 ft. bulk freighter calculated by the traditional load and buoyancy curve method and by the mean moment method.

CONDITION 1

PARTICULARS FROM BENNETT'S PAPER (REF. 12)

Light condition in still water, coal bunkers empty, hull and outfit 4700 tons, engines 300 tons, boilers 200 tons, displacement 5200 tons.

From Fig. 7: mean d = 5·65 ft., L.C.B. = 7·0 ft. f. MTI = 3740.

From Fig. 8: max. SWBM (hog) = 89150 at 50 ft. aft of midships.

SWBM (hog) = 86000 at midships.

BY MEAN MOMENT METHOD

Weight distribution

Item	*tons*	*lever*	*mt. aft*	*mt. fwd*
Hull and outfit	4700	147·0 (= ·245L)	345450	345450
Engines	300	266·0 a	79800	–
Boilers	200	235·0 a	47000	–
	5200		472250	345450

L.C.G. = 24·4 ft. aft (approx.) diff = 126800

draught, d = 5·65 ft. = ·00942L total = 817700

trim, $t = \dfrac{31\cdot4 \times 5200}{3740} = 43\frac{1}{2}$ in. mean = 408850

$= 3\cdot625$ ft. aft = ·00604L

Value of "a" (from Table 1) = ·20430

Mean buoyancy moment $= \cdot20430 \times 600 \times \dfrac{5200}{2}$

$= 318708$

Mean weight moment $= 408850$

SWBM (hog) = 90142 at midships

This BM is about 5 per cent in excess of Bennett's midship moment and 1 per cent in excess of his maximum.

CONDITION 2

PARTICULARS FROM BENNETT'S PAPER (REF. 12)

Light condition in still water (tipped for propeller examination), coal bunkers empty, fore peak and No. 1 tank filled, displacement 6610 tons, d fwd = 9 ft. $3\frac{1}{2}$ in., d aft 4 ft. $8\frac{1}{2}$ in., trim = 55 in.

From Fig. 7: L.C.B. = 6·6 ft. f, MTI = 3420.

From Fig. 9: max. SWBM (hog) = 167000 at about 23 ft. fwd of midships.

SWBM (hog) = 165000 at midships.

BY MEAN MOMENT METHOD

Weight distribution

Item	*tons*	*lever*	*mt. aft*	*mt. fwd*
Hull and outfit	4700	147·0 (= ·245L)	345450	345450
Engines	300	266·0 a	79800	–
Boilers	200	235·0 a	47000	–
No. 1 ballast tank	960	235·0 f	–	225600
Fore peak tank	450	280·0 f	–	126000
	6610		472250	697050

L.C.G. = 34·0 ft. f (approx.). diff = 224800

draught, d = 7·0 ft. = ·01167L total = 1169300

trim, $t = \dfrac{27\cdot4 \times 6610}{3420} = 53$ in. mean = 584650

take t = 55 in. = 4·583 ft. = ·00764L

Value of "a" (from Table 1) = ·21292

Mean buoyancy moment $= \cdot21292 \times 600 \times \dfrac{6610}{2}$

$= 422220$

Mean weight moment $= 584650$

SWBM (hog) = 162430 at midships.

This BM is $1\frac{1}{2}$ per cent below Bennett's midship moment and less than 3 per cent below his maximum.

CONDITION 3

PARTICULARS FROM BENNETT'S PAPER (REF. 12)

Fully and homogeneously loaded to 20 ft. even draught in still water, fuel 220 tons, cargo 13880, no ballast, displacement 19300 tons.

From Fig. 7: L.C.B. = 3·7 ft. fwd.
From Fig. 10: max. SWBM (sag) = 49540 at about 110 ft. fwd of midships.

SWBM (sag) = 42000 at midships.

BY MEAN MOMENT METHOD

Weight distribution

Item	*tons*	*lever*	*mt. aft*	*mt. fwd*
Hull and outfit	4700	147·0 (= ·245L)	345450	345450
Engines	300	266·0 a	79800	–
Boilers	200	235·0 a	47000	–
Fuel	220	215·0 a	47300	–
No. 1 Cargo hold	3700	181·0 f	–	669700
No. 2 ,, ,,	3520	71·2 f	–	250624
No. 3 ,, ,,	340	6·0 f	–	2040
No. 3 ,, ,,	2980	53·2 a	158536	–
No. 4 ,, ,,	3340	155·0 a	517700	–
	19300		1195786	1267814

L.C.G. = 3·7 ft. fwd (approx.). diff = 72028
draught = d = 20·0 ft. total = 2463600
= ·03333L mean = 1231800
trim = nil (L.C.B. = L.C.G.)
Value of "a" (from Table 1) = ·22000

$$\text{Mean buoyancy moment} = \cdot 22000 \times 600 \times \frac{19300}{2} = 1273800$$

Mean weight moment = 1231800

SWBM (sag) = 42000 at midships.

This BM is equal to Bennett's midship moment and 15 per cent below his maximum.

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Discussion

on

Mr. F. S. J. McKinlay's Paper

LONGITUDINAL STRESSES IN MODERN CANADIAN GREAT LAKES BULK FREIGHTERS

LLOYD'S REGISTER OF SHIPPING

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Discussion on Mr. F. S. J. McKinlay's Paper

LONGITUDINAL STRESSES IN MODERN CANADIAN GREAT LAKES BULK FREIGHTERS

Mr. G. Buchanan

We must congratulate the Author on producing a very good paper on this subject and it is quite evident from the mass of information which he has collected, that he has spent a tremendous amount of time and trouble on the paper. I particularly refer to the 124 conditions, ballasted and loaded, he has worked out giving the stresses in the deck and bottom for each condition.

There are many statements in the paper one would like to comment on but I think the principal item must be the Author's method of working out permissible still water stresses, and I intend to confine my remarks to this.

He has taken the f value from the Load Line Rules for sea-going ships and also the f value for lakers. He has juggled the laker f value for proportions, fresh water, etc., and has then subtracted this assumed wave stress from the sea-going total to find a permissible still water stress for lakers. I am afraid I do not agree with the method he has employed.

In the latest ore carrier Rules we have ceased to consider the fBd formula of the Load Line Rules as sacrosanct and have based our minimum scantlings on the wave B.M. with a wave the length of the ship and a height of $1 \cdot 1\sqrt{L}$.

Sadler has stated that the lake waves can be about 20 per cent greater than L/2 on a 600 ft. ship, giving a 360 ft. wave, about half the length of the 715 ft. ship with a wave height of 20 ft.

I would suggest that the best method of determining the scantlings of a laker is to take the wave B.M. with this wave and work out a wave stress which will give the section modulus of the ship as it is at present.

This will divorce the minimum scantlings required from the draught and will give a formula of the type

$$I/y = \frac{b\ L^{n} B}{\text{stress}}$$

where b is a coefficient depending on the block coefficient and the wave height,

and n depends on the wave height.

We must get entirely away from the idea that the minimum scantlings of the ship must depend on draught.

We have never agreed to the proposition that the total stress may remain constant and if the wave stress is reduced due to restricted service, then the whole of this reduction can be added to the allowed still water stress.

The Author considers that the present strength of these large lakers is just right and he has shown that the maximum still water stress in loaded or ballast conditions in service is about 4 tons per sq. in. There are higher stresses given but none are service conditions. All the higher stresses are in conditions where wave stresses do not apply.

The only statement that can be made on this basis is that 4 tons per sq. in. plus a theoretical wave stress is satisfactory, but the Author wishes to increase this permissible still water stress to about $6\frac{1}{4}$ tons per sq. in. for a 715-ft. ship. We cannot say that this 4 tons per sq. in. cannot be increased, but a jump to $6\frac{1}{4}$ tons per sq. in. seems too big a step.

When the Society obtain their new recording long-based strain gauges, we will consider putting one on board one of these large lakers and try to obtain some information on the stress they actually experience.

Mr. J. B. Davies

This is an extremely interesting paper on a type of ship which is so specialised, and has such a restricted service area, that only a few of the Author's colleagues will have much knowledge of their characteristics. It is for disseminating knowledge of this type that the Staff Association is so valuable and the Author deserves our sincere thanks for the vast amount of work he has obviously devoted to this paper.

I propose to confine my comments to the section on longitudinal bending moments and particularly to the remarks on permissible still water stresses in large lakers.

The Author points out that classification requirements are related to the Lakes Load Line Rules formula $\frac{I}{y} = fBd$ and then endeavours to make a comparison between Lakes and Ocean-going standards. He mentions that his reasoning may not receive general agreement and I must differ from him on two counts:—

(1) The assumption that longitudinal strength should vary directly as draught, and

(2) His use of L/D ratios in the comparison of Lake and Ocean f values.

When considering the first point we must endeavour to distinguish between a basis for a minimum modulus standard and the basis on which an increase above the minimum may be required. It is, I think, now generally accepted that the minimum standards for ocean-going ships should be based on the wave bending moment which is some function of length, beam and block coefficient. Now draught does not enter into this (except in a very minor way in connection with the change of block coefficient with draught) so that minimum requirements should be independent of draught, e.g. in the new Rules for bulk carriers $\frac{I}{y} = kB\ (C_b + \cdot 70)$ where k varies with L. When considering whether an increase above the minimum is required on account of design features draught may well be a parameter and in a laker, when "design features" do not vary much between ship and ship, may be an important parameter. However, on ocean-going ships this is not so and other "design features" may well have more influence than draught.

Turning now to the comparisons made by the Author between the f values for ocean-going ships and lakers, it is very difficult to agree with his reasoning that as lakers have an L/D ratio of 19 and (he assumes) the ocean-going figure is 15 then he is justified in adjusting f values in these ratios. Surely there is some confusion of thought here; any restriction on this ratio must be to maintain a certain value of inertia and thus limit the deflection.

It is suggested that any attempt at direct comparison of the requirements for lakers and ocean-going ships is liable to be misleading. While one must always consider that an ocean-going ship may, somewhere, sometime, meet a wave of about her own size, this is not necessarily so on the Lakes where geographic conditions impose a definite limit to the maximum length and height of wave. Thus the determination of minimum modulus for lakers is a unique problem and Mr. Buchanan has already spoken on this to-night.

Mr. McKinlay has given us a long and interesting paper. My criticism of one part must not be taken as lessening my appreciation of the whole.

Mr. J. W. G. THURSTON

I would like to congratulate the Author on this excellent paper. The amount of work involved must have been prodigious, but has resulted in giving a clear picture of the workings of the laker, what she is called upon to do, and why.

With regard to longitudinal strength the bases behind the reduced modulus have always been reduced corrosion and reduced wave stresses. Since the coming of the Seaway experience may show that the first of these factors should have less weight than hitherto because the vessels are now spending a portion of their time in salt water. However, it can be shown, on the basis of the Society's new Rules for bulk carriers having the class notation "Strengthened for ore cargoes", that the strength of the existing laker is equivalent, even without corrosion allowance, but I am afraid that if future designs were accepted with still water stresses of 6 to 7 tons in operational conditions as shown in Fig. 4 of the paper one would be asking for trouble.

The new rules incorporate Mr. Murray's concept, based on experience (Ref. 8), that for a ship with a high still water bending moment, the still water stress can be higher than the standard, provided the total stress is at the same time reduced. This concept is analogous to the results obtained from fatigue tests wherein a cycle of reversed stress, in our case the wave stress, is superimposed on a constant stress, the still water stress.

As a result of the fatigue tests Goodman (Ref. 15) suggested a straight line relationship between the ratio of cyclic stress to ultimate, and the ratio of constant stress to ultimate (*see* Fig. 13).

If we substitute the wave stress for the first ratio and the still water stress for the second, it is apparent that the curve obtained from the new Rules for a 710 ft. ship has the same form as Goodman suggested but falls off much more sharply with increasing constant stress. A greater still water stress, namely, 4 tons/sq. in. is permitted by the Rules in the ballast condition, presumably because the deadweight is reduced, with consequent reduction of dynamic stresses, and this curve is a little closer to the Goodman line at zero wave stress.

Sadler, referred to by the Author (Ref. 13), investigated waves on the lakes and gave the maximum observed length as 350/400 ft. with a maximum height of about 20 ft. He gave maximum stresses for various lengths of ship but unfortunately not up to a length of 710 ft. However, it is estimated that passing through waves 355 ft. long × 20 ft. high the maximum wave bending moment would be about 110,000. This gives a maximum wave stress of $\frac{110{,}000}{42{,}000} = 2{\cdot}6$ tons/sq. in. From Fig. 13 the permissible still water stress

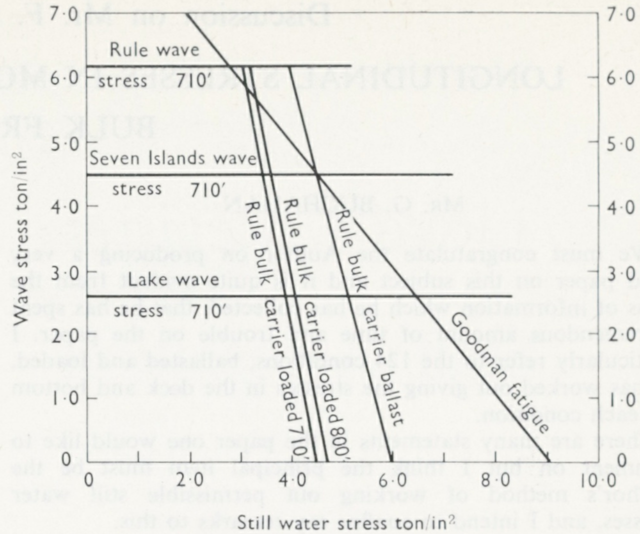


Fig. 13

corresponding to this wave stress is 3·9 tons/sq. in. or 5·2 tons/sq. in. loaded in ballast.

Since the advent of the Seaway the ships now go to Seven Islands and here the long Atlantic swell can be experienced. Assuming this could be of equal length to the ship, ordinary sea-going stresses would be experienced were it not for the fact that the height is very much reduced. From experience I would say that the height could be up to 10 ft. but is extremely unlikely to exceed that figure. These assumptions would give a wave stress of about $4\frac{1}{2}$ tons/sq. in. which from Fig. 13 corresponds to a permissible still water stress of 3·4 tons/sq. in. loaded or $4\frac{1}{2}$ tons/sq. in. in ballast.

It is observed that in all the conditions, other than still water ones, given by the Author these allowable still water stresses are not exceeded, as in Conditions 62 and 63, the deadweight is such that they rank as ballast conditions. In still water the stress could theoretically be up to the Rule operational total of just over 9 tons/sq. in. so that again the Author's miscellaneous conditions are within bounds.

From the foregoing it will be seen that it would be very unwise to conclude that the existing ships are too strong for their service and if a similar diagram be drawn for an 800 ft. ship on the basis of Rule stresses it will be seen that the only change is to shift the "Rule Bulk carrier loaded" line to the right but at the same time the wave stresses are increased, so the net effect is to give roughly the same still water stresses as in the 710 ft. ship and certainly does not justify operational still water stresses in the region of 6 to 7 tons indicated by the Author in Fig. 4. Reasonable limits for all operational conditions would appear to be $3\frac{1}{2}$ tons loaded and $4\frac{1}{2}$ tons ballasted, which conditions the existing lakers fulfil.

This analysis has been based on hogging stresses only as no large sagging stresses are evident but maximum sagging stresses could be related to the Rules in a similar manner.

Mr. R. G. LOCKHART

Whatever comments can be made on any of the contents of this paper, nothing but praise can be offered for the vast amount of work entailed in its preparation. This is at once a study of lakes and lakers past and present, a look into the

future and an endeavour to establish a basis for future trends.

In Part II the value of Table I and Tables 10, 11 and 12 for the complete picture for loading and ballasting a 710 ft. laker cannot be over-emphasised.

It is, however, an unhappy choice of method to arrive at the still water stresses.

From his comments it is unlikely that the Author was influenced by Group 7 or even much by Groups 5 and 6; it is therefore difficult to assess his desire to arrive at still water stresses so much in excess of practice. It is true that stresses cannot be dealt out like a pack of cards—2 to still, 1 to wave, etc. The Author himself states that "the margin of strength in hand is small", yet he is apparently prepared to condone stresses almost 50 per cent above those he has so carefully determined.

It is felt that any approach to this subject must be realistic even if it involves the use of hypothetical waves which the Author so disdainfully avoids.

In the case analysed by Bennett in 1928, a wave bending moment was found on the basis of a wave of $\frac{1}{2}$L and a height of 20 ft. If the length of this wave can be assumed to be increased to ·65L, with which the Author agrees, it would appear from a preliminary investigation that this amount can be expressed as a percentage of the sea-going wave bending moment. This percentage may be fairly constant for ships above 600 ft.

The proposal to continue to base the strength of lakers above 710 ft. on the fBd method also appears to be unsafe as the comparison to the sea-going ship shows a ratio which continues to fall as the length increases.

The standard of the 710 ft. ship is just under half of that of the sea-going ore ship and if this ratio continues and is associated with a percentage wave bending moment already referred to this would probably result in a still water stress of about 5–5·5 tons with a total stress of 9–9·5 tons for lakers above 710 ft.

Proportions of lakers are not extreme for their service conditions, the $\frac{L}{D}$ of 13·5 arrived at in the days of the load line formula when lengths of 400 ft. and 500 ft. were large ships. If length increases in the lake ships undoubtedly the proportions will require consideration but possibly a depth of 40 ft. in relation to a length of 800 ft. is not excessive.

It is suggested that if the proposal to increase the length of lakers to, say, 850 ft. is on the cards an early investigation should be made into the strength but not quite by the method the Author has arrived at.

In the Author's conclusion his anxiety regarding the arrangement of hatch corners is understood for undoubtedly deflections must make themselves known in these regions.

Mr. O. M. CLEMMETSEN

When the Author and I were employed in the Technical Office of the British Corporation, the Surveyor in charge was a Mr. W. I. Hay who had a wealth of experience and a fund of anecdotes about conditions in and around the Great Lakes where he had served many years in pre-war days. The Author himself has now had several years experience of conditions on the lakes during the period of considerable change due to the opening of the Seaway and it is, therefore, rather appropriate to my mind that he has chosen this subject for his paper.

In Part I of the paper one finds a very detailed account of service conditions and this will surely be a source of reference for many years. I think that the addition of a small-scale map of the area giving the principal ports from whence the various bulk cargoes are shipped would be very useful. The calculation of the stresses in 124 operating conditions for a 710 ft. laker given in Part III of the paper represents a considerable amount of work and shows in a practical way the problems confronting the operator.

With both of the foregoing parts there is very little that is controversial but such is not the case with Part II on longitudinal strength. Here the Author has given us an ingenious method based on draught of arriving at permissible stresses, but this factor is now generally conceded to have little bearing on longitudinal strength. However, I feel that regardless of the method used the final results in Fig. 4 do broadly represent current practice for lengths up to about 700 ft. If one examines the results of the calculations for loaded and ballast conditions on the typical laker in Part III, one finds that the still water stress at amidships only occasionally exceeds 4 tons/sq. in. due to limitations on deflections for negotiating canals as opposed to the $6\frac{1}{4}$ tons/sq. in. permitted for a length of 710 ft. in Fig. 4. Higher stresses approaching the latter figure can, however, be found in the ballasting sequences (Group 6), but we are assured that these are not often found in practice due to the ballasting of several tanks simultaneously. One is therefore left with the higher still water stresses due to tipping for propeller examination (Group 7), but it is not unknown for sea-going ships to be subject to much higher still water stresses during special operations in port than would be considered acceptable when at sea. While, therefore, one must be careful not to interfere with established successful practice I feel it would be preferable to have two curves for allowable still water stresses, one for the conditions when loading and ballasting at the berth and another about $1\frac{1}{2}$ tons/sq. in. less for the voyage conditions.

Making a rough calculation for a box shaped vessel with two wave crests at the quarter lengths one finds that the wave bending moment is a quarter of that which would be produced by a wave of the same height but equal to the length of the ship. It seems, therefore, that for a given modulus the wave stress in the lakes would be a quarter of that in the oceans, i.e. about $1\frac{1}{2}$ tons/sq. in. for a 700 ft. ocean-going ship but about $3\frac{1}{2}$ tons/sq. in. for the lake ship with its reduced modulus but using the same wave height. Adding this to the $6\frac{1}{4}$ tons/sq. in. still water stress proposed by the Author gives a total stress of $9\frac{3}{4}$ tons/sq. in. for the 710 ft. ship which approximates to that proposed by the Author. This would be reduced to $8\frac{1}{4}$ sq. in. for voyage conditions if the suggestion above regarding "voyage still water stress" were adopted. On the assumption that the waves on the lakes do not in fact exceed 350 ft. in length a case could be made out for no further increase in minimum section modulus for ships of over 700 ft. provided the present maximum beam is not exceeded.

Large lakers have recently been built on this side of the Atlantic and the problem arose as to whether the strength and proportions were suitable for the trans-Atlantic voyage. It was decided that the ship could only make the trip between the beginning of May and the end of September, and taking into account the reduced height of wave likely to be encountered during this period as given in weather ship reports, the reinforcement was based on obtaining a deflection, on a reduced height wave, of the same order as that of a tanker

of normal proportions on the standard wave. The ballasting of the ship for the voyage was specially approved to give an acceptable compromise between draught and stress.

For the latter the builders submitted full still water bending moment diagrams and it was observed that with the unsymmetric ballasting proposed the still water bending moment clear of amidships could exceed that at amidships. In the departure condition, the still water stress was about 2 tons/sq. in. at $\frac{1}{4}$L aft and about $\frac{3}{4}$ tons/sq. in. at amidships. It is therefore probable that in Condition No. 64 a similar situation might arise.

Mr. M. A. AHMED

The presented paper draws attention to the tremendous influence of local conditions on ship design. One is impressed by the enormity of the size of these lakers and the peculiar form and dimensional characteristics are noted.

The Author states that over large stretches of the seaway the water depth is between 30 ft. and 35 ft., with the permissible draught of 25 ft. 6 in. one could expect the problem of shallow water resistance. It is therefore assumed that these lakers require far more power to give the vessel the desired speed than that which would be necessary for the same vessel traversing a water course with greater depth. Thus, the "overpowered" engine could lead to serious trouble in the propeller aperture and the after peak by inducing severe vibratory forces from the propeller. For interest, the Author might like to quote the installed b.h.p. in one of the 710 ft. lakers.

Quite correctly the Author points to the rather academic nature of the wave bending moment calculated from the normal concepts of wave proportions. Some two years back Mr. Murray presented a paper to the Staff Association outlining a new concept in the longitudinal strength problem, popularly known as Energy Spectrum Analysis. With reasonable hope, it should be possible to use the statistical observations of the Great Lakes wind and gravity generated waves to compute a statistical measure of the wave bending moment and its probability. This statistical measure of wave bending moment could then be correlated with the actually measured stress, also obtained by a method of statistics and strain gauge technique.

The detailed description of the loss of the *Carl S. Bradley* would focus our attention on the torsional stresses on the hull girder encountered in oblique seas. Both theoretical and experimental evidence so far available indicate that deck panels having extra wide multi-openings are particularly susceptible to high torsional stresses and stress concentration peaks. It is rather surprising to see that rounded sheerstrake plating has not gained much popularity among lakers—this is considered to be standard design practice in most large tankers and bulk carriers constructed in European shipyards.

One is left amazed by the tremendous amount of ballast capacity in the lakers, and the capacity of the pumps could only highlight the problem arising out of great variations in stress levels during loading and unloading operations.

The Author is requested to comment on any statutory regulations imposed on the loading and unloading procedures. Further, the Author mentions the installation of a control console, which amongst other things gives fine control on the ballasting arrangement; it would be helpful to know if this console is a standard equipment on the lakers.

The Author is to be congratulated for giving us such an informative paper on this very special type of ship.

Mr. A. K. BUCKLE

All the strength calculations in this paper are based on a ship steaming head on into the seas; this in spite of a passing mention that worse stresses appear to occur with a quartering sea.

The practice of using direct head seas only for calculations is normal for ocean-going ships where actual waves of a length equal to the ship are possible in any ship's life, but where true wave lengths are restricted, as on the Great Lakes, it would appear that the cases considered should be those where the apparent wave length equals the ship's length, i.e. for a 440 ft. true wave length and a 620 ft. ship the ship should be set at an angle of 45° to the wave then calculating the wave bending stress.

Mr. J. MELCHIOR

Although not a member of the Staff Association, I would be pleased to make some comments on Mr. McKinlay's very comprehensive paper on Canadian Great Lakes Bulk Freighters. In Mr. McKinlay's introduction comments were invited on the riveted seams in the bilge strake. It seems to me that the reasons for their existence are very poor. Their main purpose must be to act as crack arresters in case of bottom failure. The most common cause of serious cracks in ships to-day seems to be due to low temperature brittleness of the ordinary ship steels when subjected to high tensile stresses. The temperature distribution in a ship's structure can vary considerably. The deck structure could be subjected to a temperature of −20° C. while the bottom shell would rarely have temperatures below +40° C. Looking at Tables 10, 11 and 12 in Mr. McKinlay's paper, the bottom structure is frequently subjected to rather high compressive stresses. Very few cases are tensile and then the stresses are very low. It seems to me that for the Great Lakes Bulk Freighter the riveted bottom seams are not warranted, especially with reference to prevailing loading conditions, but also considering the temperature conditions of plating in constant contact with the sea.

AUTHOR'S REPLY

I wish to thank my colleagues for the interest they have shown, for their helpful comments in the discussion and for their very generous remarks concerning the concept.

The major criticism centres on the method adopted in deriving the permissible still water stresses in Part II and a number of speakers have made this their main issue. Before proceeding to reply individually to the general comments and questions, therefore, it might be useful to enlarge on this central point.

The paper admits that the comparison of sea-going ships and lakers on the basis of "f" values and draught is "unreliable and undesirable". While this may be so, it is also considered that the theoretical trochoidal wave of about half length of ship may not be the whole answer for the large laker since,

among other things, the proportions of depth and breadth to length are both outwith sea-going experience and twisting or torque plays an active part in their service experience. There is a good case for oblique waves being a criterion by which assessment of strength could be made and I shall have more to say on this subject in the reply to Mr. Buckle. Let it be emphasized that I have nothing but admiration for the ingenuity and simplicity displayed by our present-day practice in the structural design of sea-going ships in which the minimum scantlings are based on wave bending moments but there does exist a large field of ignorance, my own included, regarding the extent to which the various conditions of service affect the large laker. Because of this uncertainty, known factors, albeit, of recent disrepute, were preferred in place of the unknown.

The general characteristics of some sea-going types evolved to such an extent in the post-war period that departure from "f" values for deriving modulus became imperative if changes were to be safely incorporated within efficient structures. No such condition exists in large lakers which in the last 60 years have remained basically unchanged so far as bending moments are concerned.

Before it was decided to base the Rule minimum scantlings of sea-going ore carriers and tankers on wave bending moments and discard "f" and "d" much knowledge from operational measurement, full-scale experiment and calculation had been collected of the moments and stresses which could be permitted in design for the expectation of safe service. We are, unfortunately, not in possession of comparable information for the laker and in the absence of this knowledge, I would suggest that it is premature to attempt to base their scantlings on present WBM theory.

Projects are afoot at the moment in Canada with the aim of providing necessary data for the better understanding of the lakers' requirements. The Canadian Authorities are investigating wave conditions in Lake Superior and the Gulf of St. Lawrence with a view to amending the Load Line Rules for Lakes and Rivers with special emphasis on strength and freeboard. The United States Coast Guard are also involved in lake studies of wave data and full-scale ship and model tests. It will probably be a few years before rewarding benefits and relaxations will materialise from this concerted effort but at least there is a movement under way which should in time bring our knowledge of lakers abreast that of the sea-going ships.

No doubt, when sufficient relevant data become available the method suggested by Mr. Buchanan will be adopted, suitably modified to conform with experience, but meantime an interim measure appears necessary to indicate the way for the transition and it was hoped the paper would be viewed in this light.

Having this in mind, Part II set out to produce a relatively quick and, if possible, sound basis for comparison of longitudinal strength of large lakers involving only SWBM which can be assessed with some certainty. The subject was treated on a *status quo* basis on the assumption that structural strength was about right, which I believe it is for the existing conditions involved in operation, and it seemed to me at this stage of our knowledge that actual intrinsic values of longitudinal stress are of slightly less importance than the causes producing these stresses. In addition, we know the method adopted to calculate the stresses has the undesirable limitation of giving values at midships only and it is obvious that the maximum bending moments and corresponding stresses for a number of the conditions investigated will be some unknown distance from amidships and greater in magnitude than the calculated values.

It was not intended therefore that the still water stresses should be used as the datum line for future design but rather that Owners and Builders might readily be able to identify safe from detrimental conditions when they carried out some of their unusual schemes of loading and ballasting and also to provide a guide to what was happening to these ships under operational conditions. There is a definite need for this guidance and I believe some measure of success has been achieved.

I do not deny the probability exists that the still water and total stresses given in the paper are too high to become the basis for structural design but for the present interim purpose they appear to be close enough to reality to form a sound basis for comparison of longitudinal strength of the hull structures of large lakers and until associated problems are better understood I aim to act for their defence.

It might also be considered that before this paper was written, we had little idea of the longitudinal stresses these ships suffered in their every-day service.

To Mr. Buchanan

I made a tentative investigation of the wave BM on the lines suggested by Mr. Buchanan and the examination was carried out with sufficient accuracy to give an indication of the results one might expect in a more complete analysis. The laker described on page 17 was used with its BP length of 710 ft. and since the wave suggested is 360 ft. in length two waves exceed the ship by 10 ft. It was assumed that a crest or trough exactly at amidships would give the greatest moments so that the double wave was allowed to overhang the ship by 5 ft. each end.

Three conditions were examined and in each case the wave profile, of true trochoidal form, was moved vertically to obtain the correct displacement and trimmed longitudinally to make the LCB and LCG coincide vertically. The Smith correction was not applied.

Condition 72, Table 11, fully loaded with a level keel draught of 24 ft. $11\frac{1}{2}$ in. appeared suitable for this experiment and in this case with a crest amidships, the hogging stress at deck due to the passing of the waves was found to be 2·21 tons per sq. in. With the trough amidships the sagging stress at keel was 2·26 tons per sq. in.

Condition 15, Table 10, a ballast condition with 7 ft. 0 in. trim and a mean draught of 15 ft. $3\frac{1}{2}$ in. was also examined. This is about the least ballast a laker would dare to carry in a heavy sea and it was found that with a crest amidships the hogging stress at deck due to the waves was 2·28 tons per sq. in. The WBM in this case was 95,979 as against 110,000 estimated by Mr. Thurston in the discussion. Since large lakers always hog in ballast there is little point in ascertaining the sagging stress due to the waves in this condition.

Incidentally, it is of interest to note that for this investigation bonjean curves were used to calculate the mean moments of buoyancy and in Condition 72 the still water mean buoyancy moment was found to be 2,505,389 and in Condition 15 the value was 1,435,999. Using the values of "a" in Table 1 the corresponding figures are 2,504,039 and 1,435,203. These values are closer than 0·1 per cent which provides confirmation on the practical accuracy of Table 1.

The hogging stresses are each about $2\frac{1}{4}$ tons per sq. in and the sagging stress slightly greater. These stresses would be something less if the Smith correction had been applied as

the influence of this correction is to reduce the WBM. There is no practical confirmation available but I would imagine that these wave stresses will be low when compared with actual service stresses. There is no way that one can assess stress by "eye" but the visible movement of large lakers in a heavy seaway implies they will at times suffer more than these theoretical figures indicate. The paper suggests the maximum wave stress suffered by a 710 ft. laker will not exceed 3·24 tons per sq. in. and it was not intended, nor was it suggested that this is a basic wave stress for design purposes because I do not think we can definitely decide the issue at this stage. There are yet conflicting factors to be considered before we wholeheartedly apply present sea-going WBM theory to large lakers.

Reverting to Mr. Bennett's paper (Ref. 12), the bending moments obtained for his Conditions 3 and 4 where he examined a 600 ft. × 65 ft. × 33 ft. laker loaded in still water and on two 300 ft. × 20 ft. waves respectively, are of interest to this discussion. In both these conditions, the vessel sagged and the total stress at keel due to the passage of the waves was 3·05 tons per sq. in. and the still water stress was 1·74 tons per sq. in. which represents a sagging wave stress of 1·31 tons per sq. in. at keel. Having regard to sizes of ship and waves this result is not inconsistent with those obtained above for the 710 ft. ship but one does have the feeling that it is too low to be made the governing criterion in assessing the scantlings of 600 ft. lakers. The wave stress value of 3·42 tons per sq. in. suggested in the paper for 600 ft. appears to have more realism.

Mr. Buchanan says, "the Author wishes to increase the permissible still water stress to about 6¼ tons per sq. in. for a 715 ft. ship"; part of this statement must be contested. I do not wish to increase the SW stress because I do not know at this stage what the basic maximum value might be. I am saying, however, that there is sufficient structural strength built into every 715 ft. laker to permit a still water stress of this magnitude to be survived in operational service. The foregoing evaluation of probable wave stresses to some extent confirms and supports this opinion.

With regard to Mr. Buchanan's remarks on total stress in association with reduced wave stresses we can agree than in sea-going ships the basic SW stress due to the standard condition can be exceeded provided the minimum section modulus is increased by a correction which is "weighted" so as to give greater relative importance to the SWBM than that of the WBM and when this is done the total stress is reduced. Our disagreement appears to be in the amount by which the addition to the SW stress should be "weighted" and whether it is necessary to reduce the total stress.

I am not aware of the theory underlying this "weighting" process in sea-going ships but for the SW stresses in lakers an empirical factor of 2/3 was introduced. It was called a hatch corner deficiency ratio but by any other name it is a "weighting" factor and was intended to have the same general effect but of reduced magnitude, as its sea-going counterpart. This factor was applied to the reduced wave stress before being added to the basic sea-going SW stress (page 14). Thus the whole of the reduction due to restricted service was not added to the allowed still water stress.

Where the SWBM in a sea-going ship is greater than the basic, the method of increasing the modulus is devised to automatically reduce the total stress and where the modulus is required to be greater than an established minimum, it is probably desirable that the total stress be reduced and Mr. Thurston throws some light on this problem. The modulus of the laker, however, has been established by successful practice and there is no desire or necessity for it to be increased. So far as total stress is concerned therefore, the two problems are different since on the one hand, because of changing patterns in elemental design a basic and established value of SW stress has been exceeded and on the other, the basic value of SW stress, whatever it might be in actual amount, is unchanged. In the case of the laker we are merely trying to apportion the complementary units of an existing whole and because of this the whole unit, represented by total stress, cannot be in jeopardy.

Also, WBM varies with b, n and B at constant length in the formula quoted by Mr. Buchanan (or, with Cb and B in Mr. Davies' contribution) and these parameters can vary greatly in sea-going ships giving widely different WBM's. In large lakers b and n (and Cb) are fixed quantities and B is virtually fixed. Consequently for all practical purposes WBM becomes constant for a given length of ship which in turn gives a constant total permissible stress. It should be noted that for lakers over about 500 ft. in length B varies between ·1L + 3 and ·1L + 5·5 ft.

A strange anomaly has arisen through this discussion; there has been strong criticism that the SW stresses presented in the paper are too high and it would appear the wave stresses are also high so that the total stresses which are their sums must likewise be in excess. And yet the total stresses are based on actual values obtained from sea-going ships in service and related to general experience rather than optimum design (Ref. 8). They also approximate to those employed by Sadler (Ref. 13). It could be concluded, therefore, that while the method of determining the permissible SW stresses is suspect and open to criticism the results cannot be far removed from the practical, and before this conclusion could be disproved, extensive practical verification would be required but the implications are clear; either the large laker cannot survive the total permissible stresses of sea-going ships, and this would indeed be odd, or, she must be too strong for her service by comparison with sea-going ships. It is, therefore, gratifying to note that the Society will consider putting one of their new recording long-based strain gauges on board one of these large lakers. Such a step must be taken before the academic essence of this discussion can be translated into practical usefulness.

To Mr. Davies

Some of Mr. Davies' queries are answered in the reply to Mr. Buchanan and further comment would be repetitious. However, his point (1) on "minimum" and "increased" modulus is worthy of additional elucidation.

In assessing the required modulus for a sea-going bulk carrier one must first determine the minimum modulus based on the WBM for the ship and secondly, if necessary, increase this minimum for SWBM. The rather wonderful thing about this imaginative and skilful process is that there is a known minimum standard. The laker has been neglected in this respect; she has no basic minimum except for the Canada Shipping Act which incorporates outmoded f and d. No one knows how light her midship scantlings could safely be made if the extremes of longitudinal stress were removed by controlled loaded and ballasted conditions. It can be said that a "high" standard of strength exists for the large laker; a standard which gives almost no trouble and provides each

large laker with sufficient strength to survive all the extremes her hazardous service inflicts upon her.

Any thoughts we might have of changing this unrestrained state can only be with a view to reducing midship scantlings in exchange for a system of mutually agreed controlled conditions of service and strict adherence to such restrictions would be extremely difficult to achieve in practice. The present mood of the Owners and operators is repugnant to this control and the time is not yet opportune for a completely scientific approach to structural design of lakers ; it may come but not at the moment.

While on this subject I would like to put on record another salient point. If our standards are to be such that they require special effort for their compliance or if restrictive practices in loading and ballasting become necessary for safety, I can say now we will court disaster. I have stressed in the paper that these ships are operated without reference to documentary procedure. Navigation on the lakes is not carried out by scientists but by artists and all art is not perfect. Each laker has a built-in factor in her strength which permits latitude in operational service ; covers up for the time when the artist makes a mistake or, more likely, is forced to sail in an adverse condition dictated by economics rather than good seamanship. Such instances are frequent and as things are at present, this safety factor must be retained. The Owners are satisfied, so also are the Underwriters.

With regard to point (2) I am in agreement with Mr. Davies that where the length to depth ratio is great, deflection limited by inertia becomes the criterion in strength but inertia is a function of modulus and so deflection can be maintained by proper adjustment to modulus. The only mandatory strength requirement for large lakers is contained in the Lakes and Rivers Load Line Rules and Rule 44 (ii) on longitudinal strength has a paragraph which reads:—

"Where the length of ship is in excess of 600 ft. and the ratio of L over D exceeds 19 the factor f is to be increased to the satisfaction of the Board."

This paragraph, which is also part of the Canada Shipping Act, 1934, assumes that deflection can be controlled by modulus.

No one has yet devised a suitable basic standard for deflection. Also, unfortunately, when the paper was written there was no sure guide available to me on the strength and proportions of large sea-going bulk carriers and it might have been considered presumptuous to quote the Society's advance tentative proposal for minimum scantlings which was in my possession at that time. Actually, I was left with very little choice in the matter but I do agree there are undesirable shortcomings inherent in the f factor method adopted.

If we take a 700 ft. $\times$ 75 ft. $\times$ 39 ft. ship having a Cb of ·88 at ·06L draught and a summer draught of 26 ft. 0 in. we find that the minimum modulus required by the Society's latest published Rules (1964) for bulk carriers is 88,733 sq. in. ft. for sea-going service. As a laker this ship would require a modulus of 41,395 and from this we find that the laker strength is 46·65 per cent of the sea-going requirement for this length. By the method adopted in the paper, and which has been so ardently criticised, the ratio obtained for comparison purposes in Fig. 3 was 50·2 per cent at 700 ft. For the purpose intended this represents 7·6 per cent on the safe side and this safety factor is incorporated in the increase made to the still water stress for the reduced wave stress due to restricted service. Similar comparisons made for other lengths follow the same pattern but the values for lake ships each include draught and, of course, the differences in strength vary with this factor. However, sufficient corroboration is obtained by these comparisons to substantiate the results presented in the paper and amply justify the use made of them.

To Mr. Thurston

We can usually rely on Mr. Thurston to produce a comprehensive and balanced study of any point he wishes to make and his present valuable contribution based on his wide knowledge of lakers is no exception.

His central criticism is essentially the same as was made by Mr. Buchanan that where the basic SW stress is exceeded the total stress must be reduced but he has brought to the discussion information which supports my contention in this matter.

Mr. Thurston arrives at a maximum wave stress of 2·6 tons per sq. in. in his example and developing this by the sea-going method obtains a permissible SW stress of 3·9 tons per sq. in. This wave stress is slightly higher than obtained in the reply to Mr. Buchanan but no matter, in effect, he is saying that the total stress of over 9 tons per sq. in. should be reduced to $2{\cdot}6 + 3{\cdot}9 = 6{\cdot}5$ tons per sq. in.; the reduction is equal to the amount to be met or, alternatively, twice the anticipated maximum wave stress is in reserve to encounter the wave. Viewed from this angle the "weighting" of the SWBM, already referred to in the reply to Mr. Buchanan appears to be excessive when applied to lakers.

Incidentally, the straight line relationship in Fig. 13, after Goodman, shows that the simple summation of cyclic and constant stress should always give total stress and this is not analogous to the Society's method of adding the stresses. This in fact, is why the slope of the rule bulk carrier curves are steeper, and I consider are too steep, when applied to large lakers. Goodman's line by definition has a 1 : 1 slope whereas those for rule bulk carriers are steeper than 3 : 1. The values of stresses presented in the paper follow the Goodman line exactly and in addition the suggested laker SW stresses are "weighted" by the 2/3 factor used in transposing the reduced wave stresses.

As Mr. Thurston says the swell in the Seven Islands area of the River St. Lawrence probably produces the most severe wave conditions experienced by large lakers. Assessment of wave stresses from assumed data may be misleading and a study of waves in this area is being undertaken by the Department of Transport who might soon put us in a position to understand the problem better. In view of the Owners experience of service to Seven Islands since the Seaway opened it will be difficult to convince them that the conditions are more severe than in Lake Superior especially when it is remembered the ships are substantially the same as were developed for the Upper Lakes only. Whatever figures are produced and I am assuming they will be adverse, the argument will not be convincing. It would be much better to await authenticated information on Gulf conditions rather than enter into this controversial subject so ill prepared.

Factors influencing corrosion have changed for the large laker operating to the mouth of the River St. Lawrence and this is a good point raised by Mr. Thurston. This could well have a bearing on the future strength of these ships and some Owners are now finding it necessary to coat ballast tanks which previously required no protective treatment.

To Mr. Lockhart

Mr. Lockhart, in reading thus far, will have become more acutely aware that my views on the hypothetical wave are both sympathetic and cautious lest we should prematurely hail it as the elixir of the laker. I have no doubt that some basis involving wave theory can be evolved for the laker but much more information and data than we have now will be required before the quality of realism can be made an ingredient. As Mr. Davies has said, the determination of minimum modulus for lakers is a unique problem.

With regard to the magnitude of the suggested SW stresses, the conditions examined were based on the operations of two of the better lake ship companies and while they substantially represent normal practice, irregularities occur in service outside the general code and these appear to produce more severe stresses on the structure than the stresses recorded in the paper. For various reasons, I failed to obtain sufficient data from any of the cases known to me, to make a reliable estimate of the stresses produced by the infraction but none of the ships was lost or damaged. Even without direct proof it is evident that still water stresses something higher than those examined could be and have been survived. With our present limited knowledge assessment of the actual safe limit of still water stress for voyages on the lakes is to some extent a matter of intuition and I believe the suggested permissible SW stresses are in keeping with survived unorthodox events.

The lake ship owner knows well the economic value of the large ship and he will pursue relentlessly his vision of the super laker. There is much talk and even some action towards developing suitable designs for ships up to 950 ft. in length. Because of restricted harbour facilities making breadth, depth and draught unpredictable factors in the designs, the building of an actual ship of this size is not imminent but should harbour lay-outs permit and operation be confined to one lake then we might see a super laker trading between Canada and the States. It is fair to say, however, that the time has arrived when the Society should be prepared to state their acceptable standard for these ships and as Mr. Lockhart suggests an early start should be made into this strength problem.

To Mr. Clemmetsen

It is opportune that Mr. Clemmetsen should preface his remarks with a word of praise for Mr. Wm. I. Hay, Assistant Chief Ship Surveyor to the British Corporation, whose death in 1949 robbed the Unified Society of a wealth of experience in Canadian Lake ship practice and structural design. We are now in a different world of concepts and symbols but the end product is the same and those of us fortunate enough to have had the benefit of his tuition in our more formative years in the industry will remain indebted to him for the basic knowledge of ships gleaned under his guidance.

At Mr. Clemmetsen's suggestion and for the benefit of those unfamiliar with the geographical lay-out of the lakers' field of operations an outline map of the Great Lakes and River St. Lawrence showing some of the principal ports is given in Fig. 14.

It is rather pleasant for the Author to read Mr. Clemmetsen's contribution to the discussion as he appears to be in complete accord with the results. It is with some reluctance therefore that I question the necessity for having a "voyage still water stress" of about $1\frac{1}{2}$ tons per sq. in. less than the total stress. Surely the $6\frac{1}{4}$ tons per sq. in. which has been severely criticised for being too great for the 710 ft. laker would be ample for a "voyage still water stress" without introducing the $8\frac{1}{4}$ tons per sq. in. he suggests.

In imitation of sea-going requirements, there might be a case for having a basic loaded still water stress about $1\frac{1}{2}$ tons per sq. in. less than that for the basic ballasted SW stress when we ultimately produce structural designs by calculation comparable in procedure to the present sea-going system but it is premature to say whether this would be justifiable. The *Carl D. Bradley* sank in a ballast condition.

Mr. Clemmetsen refers to recent large lakers built in Europe and this is a delicate point for certain Canadian interests. Diplomacy decreed that no reference be made to these foreign built ships which were purposely omitted from the paper and there is no gainful reason to break the silence now.

To Mr. Ahmed

Speeds are controlled by various authorities where width and depth of channel are restricted and the Seaway Authority stipulates a maximum speed of 6 miles per hour (5·2 knots) over the bottom for vessels exceeding 260 ft. in length moving in any Seaway canal. There are also rules regarding overtaking other ships and passing moored vessels and equipment in canals. In general, therefore, maximum power is only required in open water.

Nevertheless, large lakers which all have single screw propulsion, suffer vibratory effects from shallow water resistance and experience in their operation has shown that tanker type structural arrangements are necessary aft of the hold space. Vibratory shudders are usually sufficiently in evidence to indicate when a vessel is traversing shallow water without having to consult a chart to confirm the depth. To minimise this effect of propeller incited vibrations numerous web frames and stringers are fitted in the engine room and after peak. There is no "rule" requirement for this stiffening, experience and the "last ship" being by far the best guides to success. Of some 15 large Canadian lakers built in the last few years only two, and these were turbine driven sister ships, gave uncomfortable movements when at full power and additional stiffening in the after peak modified the condition to the operators satisfaction.

Since about 1952 steam turbine machinery predominated as the main propulsive unit in Canadian-built lakers over 500 ft. in length. These units have evolved in the 710 ft. ships to machinery having HP and LP rotors developing 9,000 b.h.p. at a propeller speed of about 110 r.p.m. and capable in emergency of 10 per cent overload. In open waters maximum speeds slightly more than 17 m.p.h. ($14\frac{3}{4}$ knots) can be obtained with these engines. It should be mentioned that the unit of speed used on the lakes is invariably miles per hour and emphasises the existence of aloofness to the traditions and conventions of sea-going practice.

More recently, multi-diesel arrangements have found favour and while these units are in a minority there appears to be growing interest in this type of propulsion. The latest arrangement in the 710 ft. ships is to fit four opposed piston non-reversing 12-cylinder units each developing 2,000 b.h.p. at 720 r.p.m. These engines are arranged, two forward and two aft, each with drives into a central gear box. In addition, two smaller diesels are arranged transversely, lp and ls, with drives into the gear box to act as boosters and provide an additional 1,500 b.h.p. Thus the total developed b.h.p. is 9,500 at a propeller speed of about 115 r.p.m. giving a maximum speed in excess of 17 m.p.h. in open water.

The primary function of the two small wing diesels is to act as generators for powering the side thruster at the forward end of the ship for manœuvring in confined waters when full propulsive power is not required and also for powering self-unloading equipment which will only be operated in harbour.

Some politics are involved which we need not go into but the practical benefits of the multi-diesel are in greater control over constantly fluctuating power requirements and in reduced engine room personnel with increased automated control. Three maximum-sized multi-diesel lakers are at present under construction. One of these will have fully automated propulsion controlled from the bridge and will be fitted with an automatically controlled variable pitch propeller.

Large single unit oil engines have not yet been fitted in any large laker built in Canada but the first one will be constructed this year having a six cylinder medium oil engine developing 9,600 b.h.p. to drive a controllable pitch propeller.

As already referred to, the Canadian and United States Authorities are proceeding with comprehensive investigations of the lakers' service problems. Included in these programmes is an examination of wave conditions in Lake Superior and the Gulf of St. Lawrence involving the following branches of the Canadian Government: Marine Regulations, Meteorological Services, Marine Works and Marine Operations of the Department of Transport and the Oceanographic Groups of the Department of Mines and Technical Surveys. The Bedford Institute of Oceanography at Halifax, N.S., has recently prepared a detailed energy spectrum analysis of weather conditions in these areas based on information and data extracted from weather charts covering a five years' period; this report is not yet published. It is planned that wave recorders will be set up in Lake Superior and the Lower St. Lawrence River in an attempt to co-ordinate the hindcast studies obtained from energy spectrum analyses so that more precise data may be obtained. Strain gauging of ships in service has been discussed and is beginning to appear imminent.

There is, therefore, reasonable hope that by the use of statistical observations of Great Lakes wind and gravity generated waves, a statistical measure and probability of wave bending moments might be computed. Correlation to actual ships in service also seems feasible but this final goal is yet some time off in the future.

The question of rounded sheerstrakes has been critically examined by lake ship Owners and their findings are against it; their reasons are many and sound. The transverse extent of hatchway coamings is great in these ships and with the travelling deck crane, the rails for which are shown in Fig. 5, passageway at the ships' sides is limited. With a rounded sheerstrake bollards and guard rails would require to be placed about 2 ft. inboard of their present positions further reducing the free space. When in canal approaches and hogging due to high sun temperatures in the load condition, hosing of the deck and containment of some free water on this non-sheered level deck can be facilitated with the present standing edge of the sheerstrake and with the scuppers temporarily plugged. Spar deck plating amidships is $1\frac{3}{8}$ in. to $1\frac{1}{2}$ in. in thickness and is about the limit which can be economically worked by the shipyards. With the rounded sheerstrake effective width of deck material is reduced and the area of the stringer angle is lost necessitating thicker deck and sheerstrake plating which would be undesirable. Apart from quay wall damages sheerstrakes never give any trouble. These are some of the reasons for the present arrangement and when we consider there is no real argument in favour of rounded sheerstrakes it ceases to be surprising that these have not been adopted in large lakers. If, instead, something could be done to hatchway corners to more efficiently combat the high stress concentrations in these extra wide multi-opening deck panels referred to by Mr. Ahmed we would be contributing a beneficial service.

There is no statutory regulation imposed on loading or unloading procedures, even for grain, except in so far as the safety of personnel and cargo are concerned. Both operations, and we are considering the longitudinal stresses induced in the ship, are entirely the responsibility of the Master and his chief deck officer. Their problem resolves itself into two basic criteria which are draughts and time. Various authorities have limits on draught for areas under their control and the Owners want the maximum permissible draught in the shortest possible time. Satisfactory attainment of these two factors is derived entirely from personal experience.

In recent years the extent of automation in the control of various operational procedures on board ship has increased enormously and it can be said that the console in the engine room controlling all phases of ballasting is now standard equipment.

To Mr. Buckle

I am glad that Mr. Buckle has raised the question of the effect of a quartering sea on structural design as the opportunity is now available to enlarge on this point.

The traditional mode of strength comparisons for ships in a seaway, since the days of Rankine and Froude, assumes the wave crest at right angles to the ship's length and for normal sea-going proportions has proved eminently successful. Large lakers are narrow and shallow for their lengths by comparison with sea-going ships putting emphasis on combined deflection and twisting of the hull and these dimensional characteristics are probably more important than actual wave length. Even under moderate weather conditions visible movement of the hull can be present and, for basic structural design, it becomes obvious that consideration must be given to breadth and depth ratios and to the fact that maximum twisting is produced by oblique seas.

The effect of oblique waves on long slender ships has been known for many years and considered as a probable method of assessing laker scantlings. In the reply to the discussion on a paper (Ref. 13) given in 1959, the Author, Mr. Mack Earle, quoted an investigation made by Lorenz Hansen (Ref. 14) in which longitudinal stresses were calculated to be several times the normal wave condition when a 600 ft. laker is suspended on 300 ft. by 20 ft. waves set at an angle of 45° to the hull and Mr. Earle pointed out that this increase in stress would be further amplified by secondary racking and transverse stresses. Unfortunately, the firm who published Mr. Hansen's treatise appears to have gone out of business and I have been unable to obtain a copy of this article.

The problem is intriguing but the enormity of the calculation required for its solution has obviously discouraged investigation. When suspended angularly on two or more waves the ship will roll and it may be that maximum stress coincides with maximum heel. Therefore, in addition to so positioning the wave profile to obtain proper trim and displacement it is necessary to adjust for varying degrees of list and since at any given transverse section of the ship the water surface is not symmetrical port and starboard the initial calculation for equilibrium is quite formidable. Also, the

Smith correction should be applied to be consistent with ocean practice.

It is not known what angle of wave, in association with specific lengths of ship and wave, will give the maximum condition of stress. Therefore, for a given length of ship varying lengths of wave and varying angles of wave to ship would require to be examined. Obtaining the value and position of the maximum principal stress induced by twisting in association with the longitudinal still water and wave stresses would be a complex operation.

This is a preliminary statement of the problem but it should be enough to convince Mr. Buckle that its solution for even one ship would be quite a feat. With electronic computers it may be possible to devise a programme to determine the severity of this pattern of waves on a laker hull but it is patently recognisable that without the use of computers calculations of this nature would be excessively time consuming and economically prohibitive.

The pursuit of perfection is a laborious business but the difficulties involved do not detract from the opinion that this method of attack on the laker strength problem might be the ultimate solution.

To Mr. Melchoir

Quite soon after my initiation into the mysteries of lake ship operation, I formed the opinion that the stresses in the bottom structure of large lakers were of secondary importance to those in the deck and that the longitudinal riveted seams at the bilge fulfilled little practical purpose. Subsequent experience strengthened this opinion.

With regard to these seams, we are, as a general statement, concerned with high tensile stresses and lack of notch toughness in association with relatively thick mild steel plating and low temperatures. Usually, sea-going tankers, ore carriers and bulk carriers sag when uniformly loaded producing tensile stresses in the bottom and these stresses are often of such magnitude that it is necessary to increase the basic modulus to provide sufficient longitudinal strength to accommodate them. Under such conditions the fitting of riveted seams and Grade D quality steel appear to be reasonable and practical precautions to take.

In the large laker tensile stresses in the bottom are comparatively low. When in operation the air temperature would rarely be much below zero degrees Centigrade and the $-20°$ C. quoted by Mr. Melchior could only be in winter when the ships are laid up. Then the bottom is either in compression if the ship is winterised with no cargo on board or a small tensile stress might be present if loaded with winter storage-grain. The bilge strake is ·81 in. thick with adjacent bottom plating ·75 in. thick and these thicknesses are on the fringe in which operational low temperature brittleness is adversely present in ordinary Grade A quality mild steel. In these circumstances, therefore, I must admit to personal agreement with Mr. Melchior in his views on the omission of these riveted seams at the bilge. Fortunately, the onus of decision rests with others and I am not well enough versed in metallurgy of steels to overpersuade my opinion. Separate relevant facts can only be presented as they appear to be in the hope that they may be considered in relation to the complete subject and utilised towards advancement of our knowledge.

Finally, I would like to add that the quantity and quality of the discussion has been gratifying. I greatly appreciate the trouble taken by contributors to the discussion to formulate their comments and it is hoped the explanations in the replies will be found satisfactory to them.

ADDITIONAL REFERENCES

13. Earle, M. Mack: "The Conversion of T2 Tankers for Great Lakes and Seaway Service", S.N.A.M.E., Great Lakes Section, September, 1959.

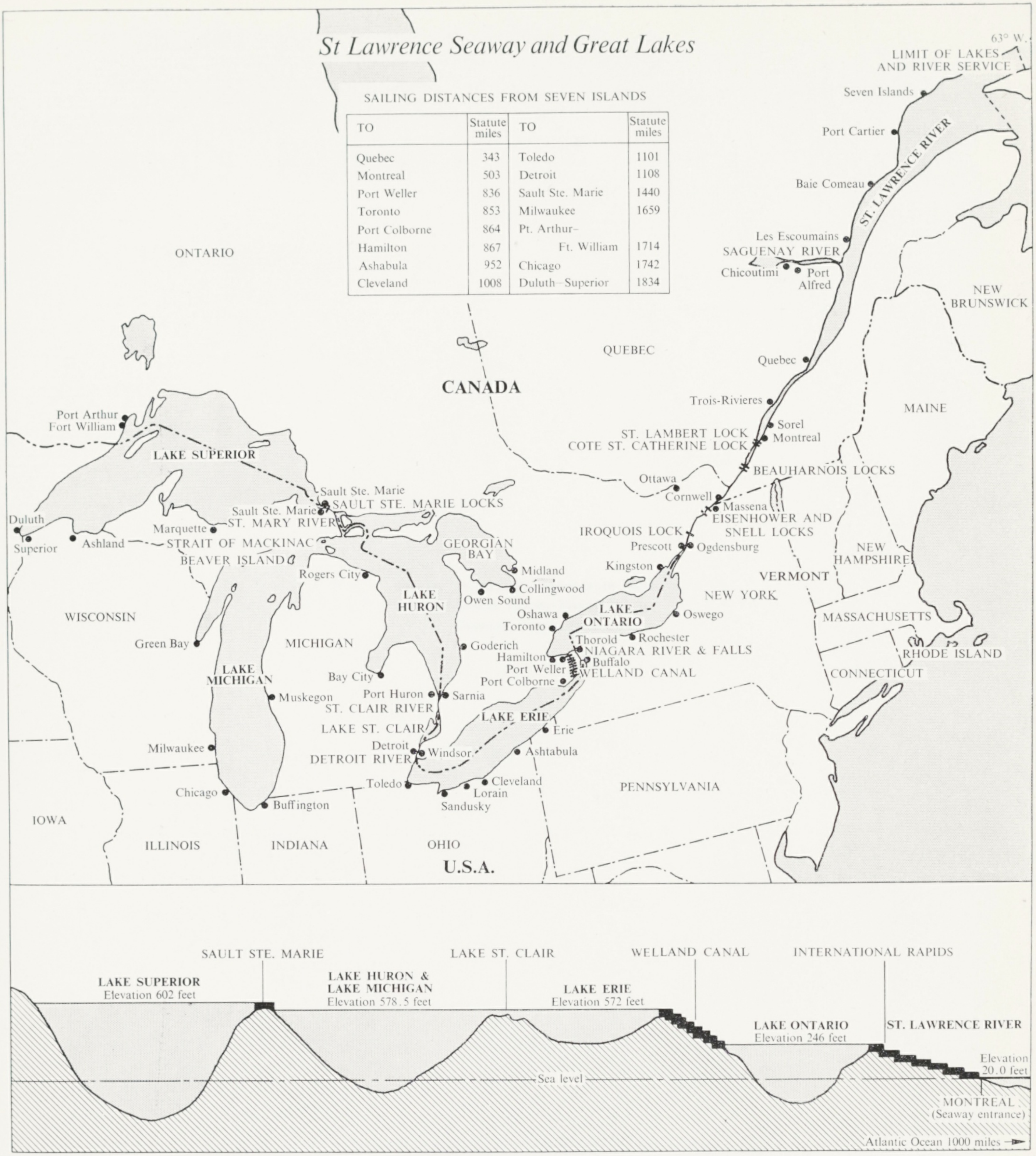
14. Hansen, L.: "Waves", Marine Engineering and Shipping Age, November, 1930.

15. Morley: Strength of Materials, Chapter III, Article 50.

ADDITIONAL ILLUSTRATIONS

Fig. 13 Curves of Stresses.

Fig. 14 Map of Great Lakes and River St. Lawrence.



TO	Statute miles	TO	Statute miles
Quebec	343	Toledo	1101
Montreal	503	Detroit	1108
Port Weller	836	Sault Ste. Marie	1440
Toronto	853	Milwaukee	1659
Port Colborne	864	Pt. Arthur– Ft. William	1714
Hamilton	867		
Ashabula	952	Chicago	1742
Cleveland	1008	Duluth–Superior	1834

FIG. 14

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STRAIN GAUGE TECHNIQUES

by

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STRAIN GAUGE TECHNIQUES

By A. J. COGMAN

In 1953 the writer presented a paper entitled "Electric Resistance Strain Gauges".

Since that time the use of these devices has increased to an unforeseeable extent in most fields of engineering. The increase in use has been accompanied by an equally notable improvement in the technique of application and instrumentation of strain gauges and this paper attempts to convey the broad outline of present-day strain gauging, with as little as possible reiteration of the basic principles given in the former publication.

In a relatively short paper, it is not possible to give details of every type of application to which strain gauges may be put; it is hoped, however, that the examples used, which are all typical of the type of work carried out by the Society's Engineering Investigation Dept., will prove of interest and may suggest further fields in which what has become a very versatile engineering tool may be employed.

It may be mentioned in passing that it has been estimated by an American authority, that, at the present time, as many strain measurements are made in one day as were made in ten years prior to the invention of the resistance strain gauge.

As is probably now well known, the wire strain gauge was originally devised by Simmons and Ruge of the U.S.A. in about 1939. Subsequent development that occurred was rapid, probably due to the onset of the Second World War, and the wire strain gauge, which soon entirely replaced the earlier carbon element gauge already in use in the aircraft industry, found ready application in the many dynamic and static strain problems present.

In the U.K. the development of the wire strain gauge commenced in the early 1940's, and while certain pioneering work was effected, it proceeded initially along lines limited by the available instrumentation in this country at that time, and it is reasonably correct to say that the strain gauge instrumentation at present practised here is more influenced by American than by local or European tendencies. The foil strain gauge, the development of which has proceeded almost entirely since the appearance of the writer's last paper, although a British invention, appears to have been equally, if not to a greater extent, influenced by the U.S.A., where it now appears to be replacing the wire gauge to all intents and purposes.

It may well be, however, that even this later development will be rendered obsolete by the newer semi-conductor strain gauge, which by virtue of its higher sensitivity will operate with much simpler instrumentation.

For the future, it is as impossible to predict developments as it would have been ten years ago to predict the present trend; it seems likely, however, that semi-conductor gauges, operating without the use of electronic amplification, may well replace other forms of strain gauge in the foreseeable future. Also the use of strain gauges in permanent installations for weighing, load measurement, etc., which is now accepted in many quarters, will certainly increase.

Basic Characteristics of a Resistance Strain Gauge

Kelvin, in 1856 discovered that when a wire was subjected to tension, its resistance increased in proportion to its load and consequent extension. He devised a scheme which would, in fact, measure, albeit rather crudely, load in the wire by means of the accompanying electric resistance change. Apart from recording this data, no further attention was paid to this phenomenon at that time.

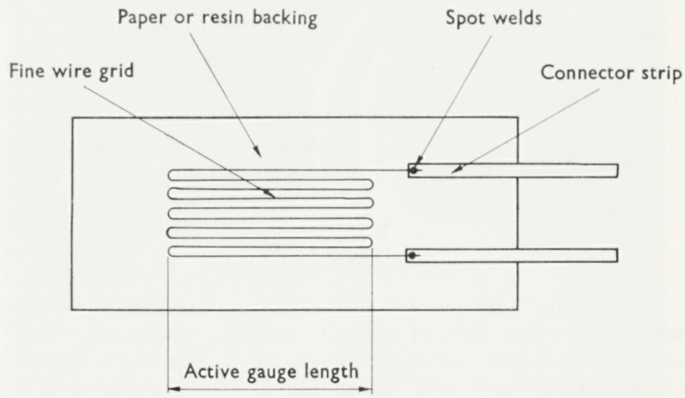


FIG. 1

Grid gauge

In about 1938, Simmons and Ruge applied the principle of bonding a fine wire to a metal surface to measure mechanical deformation of the surface in terms of resistance change of the fine wire. In order to provide a sufficiently large resistance to facilitate accurate measurement, and yet to localise the measuring effect over a small area, the wire was applied in the form of a grid to the surface, from which it was insulated by means of a thin layer of paper. The "grid" type of gauge (*see* Fig. 1) held the field for many years, especially in the U.S.A., which was its country of origin. The earliest manufactured gauges were of this pattern and, incorporating as it does, only one layer of insulant between the gauge element and cement, probably gained its popularity by the excellent repetition accuracy obtainable in large scale production. At about the same time that the grid type was being developed in the U.S.A., the helical type was becoming available in the U.K. This gauge (*see* Fig. 2) was originally made by winding a fine wire coil on a thin paper former, flattening the completed coil and former and embedding the whole in a paper-bakelite matrix provided with suitable end connections for the wire.

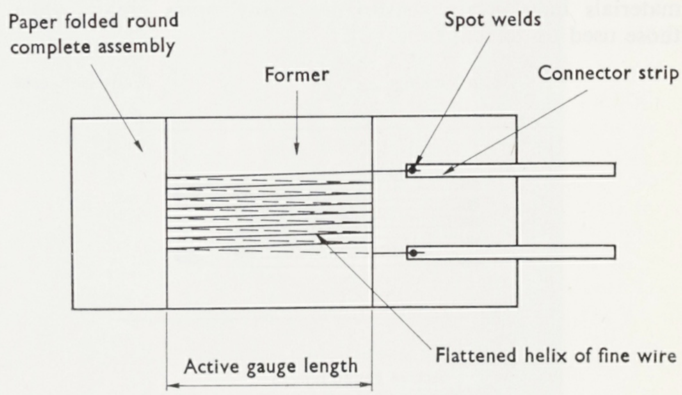


FIG. 2

Helical gauge

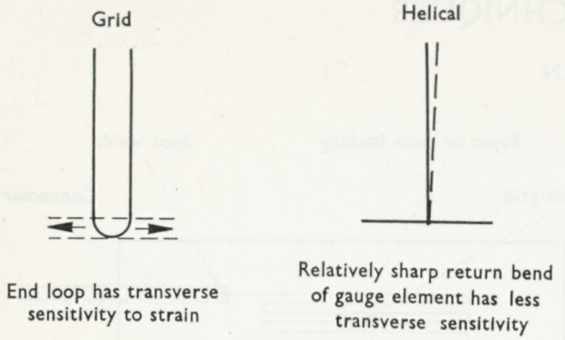


FIG. 3

Comparison of gauge ends

The greater number of active layers in this type of gauge rendered it more difficult to apply than the grid gauge when using the cements commonly available in the early 1940's. Consequently the helical (or in the U.S.A. "wrap-around") gauge was slower to be adopted but in the early development days it scored over the grid type when short gauge lengths were required, as it presents a lower "cross sensitivity", i.e. response to transverse strain, than does the grid gauge. (Fig. 3 illustrates this comparison.)

In the early 1950's a strain gauge was developed by Saunders-Roe which was based on the printed circuit technique being adopted for radio and electronic construction at the time. In its early forms the foil strain gauge, as it became known, had certain limitations, but its undeniable advantages quickly led to its adoption on a large scale, particularly in the U.S.A.

The gauge consists of a piece of thin foil which is etched to the shape shown in Fig. 4, the proportions and dimensions being varied to suit particular requirements. The etched foil is usually attached to an epoxy resin backing film for use at normal temperatures but at elevated temperatures it may be bonded directly to the specimen using a ceramic cement which both acts as an insulator and transmits the strain to the foil. As may be seen from Fig. 4, it is possible to form the shape of the foil "grid" so that cross sensitivity is minimised.

The initial drawback of the foil gauge was its low resistance, which was usually less than 50 ohms in early examples; modern production techniques have raised this to 120–150 ohms for gauges down to as little as ·1 in. active length.

In the late 1950's, experiments with semi-conductor materials such as germanium and silicon, showed that these materials have strain sensitivities many times greater than those used up to that time.

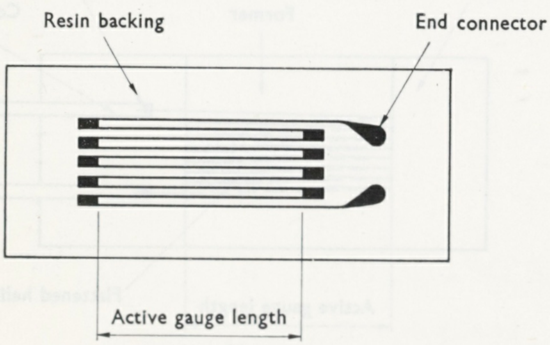


FIG. 4

Foil strain gauge

Coupled with this property is the fact that the materials may have a resistivity which is also higher, enabling a small strain gauge to be produced having satisfactory electrical characteristics and such a high sensitivity that the use of amplification between the gauge and the reading device is often unnecessary.

This is a very real advantage when size and weight considerations are of importance, and as it eliminates a link in the measurement chain it is also a contribution to greater reliability and accuracy.

Attachment of Strain Gauges

The first stage in any strain measurement problem, after positioning of the gauges has been decided from mechanical consideration, is to attach the strain gauges to the surface of the test piece. As may be readily appreciated, the gauge has to perform the function of responding to the deformation of the test material as exactly as possible and this implies that the adhesion of the gauge must be as strong as possible and

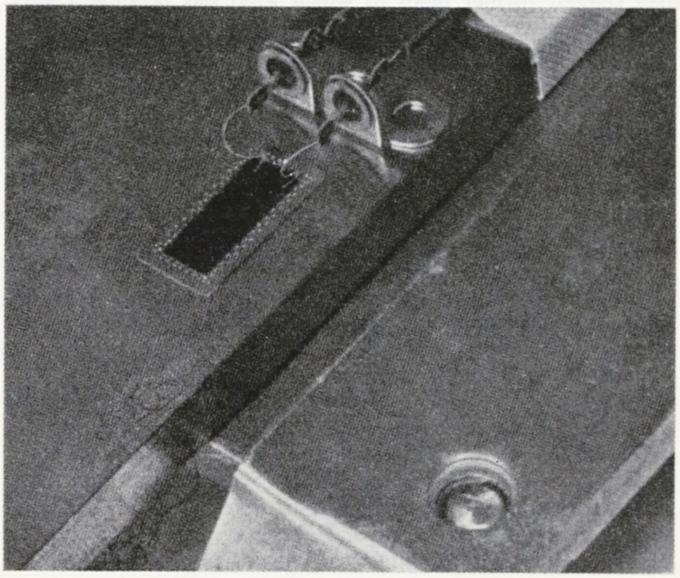


FIG. 5

Weldable strain gauge installed

also uniform over the sensitive area of the gauge. Surface preparation for strain gauge measurement must therefore provide a smooth surface which is free from dirt and grease. In the case of steel test surfaces mill scale must be removed, preferably with a disc sander or similar tool, leaving a metal surface which is bright and free from pits and deep score marks. However, too highly polished a surface may reduce adhesion undesirably; a very good surface may be obtained by sand-blasting. Other metals such as cast iron, aluminium, aluminium alloys may be similarly treated although hand emery papering is often better on the softer materials if the area to be prepared is not large.

Cements used with strain gauges have developed considerably in the past ten years.

Whereas ten years ago the use of cellulose cements was widespread in the attachment of strain gauges, they were somewhat unreliable, particularly in the humid conditions met with in marine surroundings. Cellulose cements are susceptible to moisture, and as their hardening depends on the

evaporation of a solvent, they are very slow to set at low ambient temperatures. They are now (or should be) completely obsolete for this purpose.

Thermo-setting cements, such as the "Bakelite" phenol-formaldehyde type, were a step forward except that, as their satisfactory hardening usually depended on baking to about 150° C. their use in marine applications was limited to those parts which could readily be raised to such temperatures. About 1953 a range of synthetic resins known as epoxides was introduced in forms applicable to the attachment of strain gauges. These cements cure at room or slightly elevated temperatures in about 15 hours and possess excellent resistance to water ; as a means of transmitting the strain to the gauge they are at present unsurpassed.

A few years later a cement consisting of a polyester resin base was introduced, which is capable of hardening in about 15 minutes and transmitting strain after about one hour at 20° C. While its strain transmission properties are slightly inferior in some ways to the epoxides, the advantage of quick setting is one which has made cements of this class extremely popular for all branches of normal temperature strain measurement.

Another cement which has been successfully used for strain gauge work in recent years is "Eastman 910", a cyano-acrylate cement which hardens when pressed between two surfaces. Cements of this kind harden at a phenomenally rapid rate, in fact in the case of "Eastman 910" about 2–3 seconds. This is of value when a rapid check is necessary from a few strain gauges, but the high cost of the cement usually precludes its use on large-scale test work, and its long-term stability is questionable.

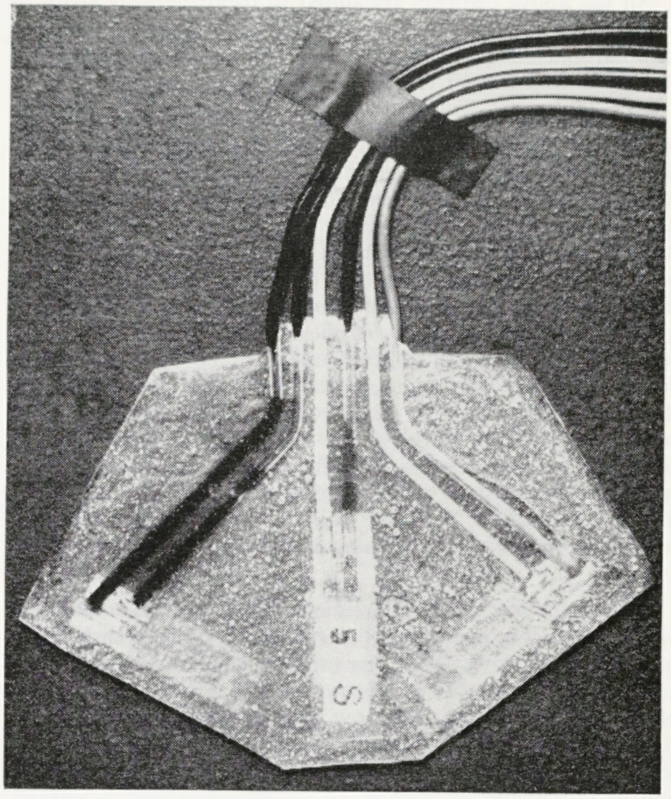


FIG. 6

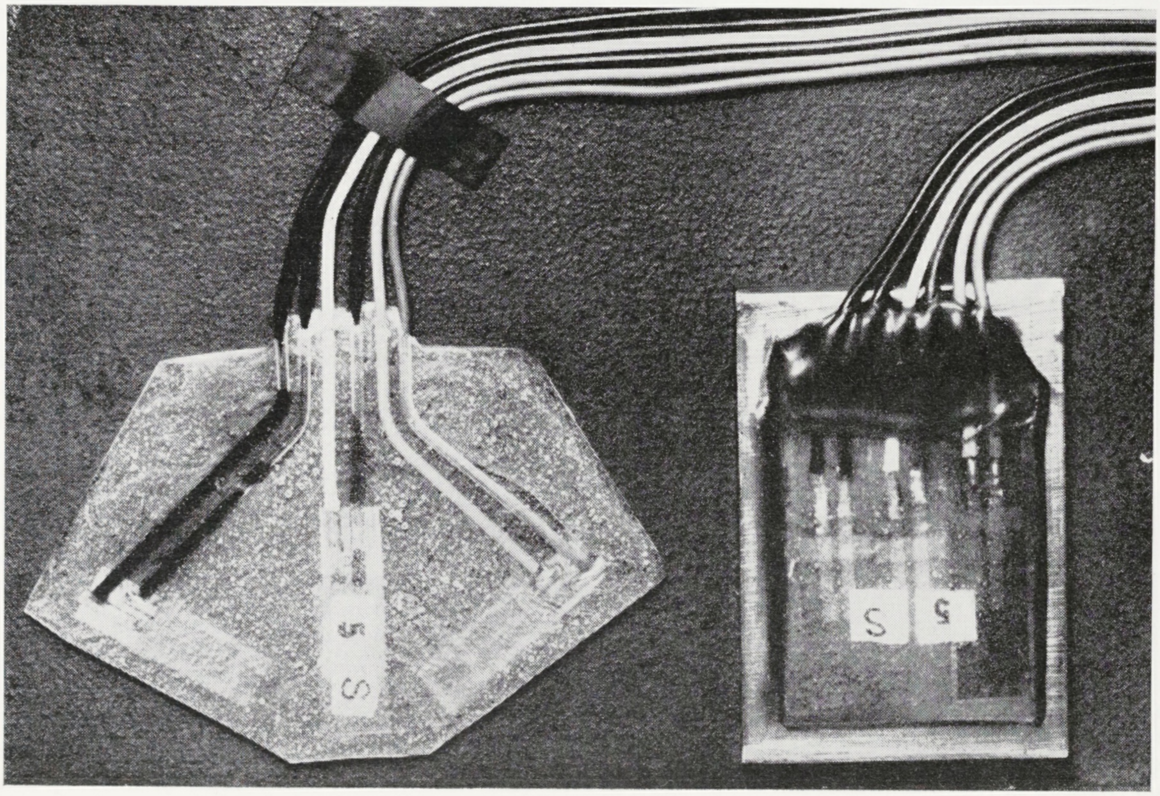


FIG. 7

All the above cements are suited to measurements at up to a maximum of 100° C. or slightly higher. Above this temperature carbonisation may take place resulting in destruction of the gauge and loss of the gauge-to-metal bond.

"Bakelite" cements of some kinds will operate up to about 150° C. before failing. Above these temperatures it is necessary to use different cements; synthetic resin cements are available, generally epoxides, which will operate up to about 250° C.; above this temperature it is at present necessary to use ceramic cements. Both kinds usually require baking to a high temperature (at least that at which the gauge will be used), before they are usable and the baking cycle is frequently quite complex and involves close control of temperature. As this may be difficult under other than laboratory conditions, a technique has been developed which enables the gauge cementing to be effected in the laboratory and subsequently the prepared gauge may be quickly installed on site. The method consists of cementing the gauge to a thin piece of stainless steel, usually about ·005 in. (0·12 mm.) thick. This is then taken, complete with its strain gauge, and spot welded to the test surface using a miniature spot welder (*see* Fig. 5).

This method of gauge attachment has proved of inestimable value in attaching gauges to internal parts of heat exchangers in the nuclear power field; its rapidity and the fact that little surface preparation is necessary may well extend its use to other fields including, e.g. low temperature work on ships' structures and other field applications where low ambient temperatures and high humidity render other methods difficult or impossible.

Its adherence to a linear strain resistance-change characteristic is as good over a normal working range, say up to strains of 1000×10^{-6}, as a conventionally attached gauge. The spot weld energy required for stainless steel of ·005 in. thickness is about 12–14 watt/seconds on an electrode of ·05 in. diameter.

The above account of strain gauge attachment is by no means complete; it would be beyond the scope of a short paper to cover more than a few methods and techniques in this field are also constantly changing.

Protection against environmental conditions

As with any electrical measurement technique involving great precision, the exclusion of water and other detrimental matter likely to affect the stability of electrical resistance is of paramount importance.

It is usual therefore to provide a protective covering of a material which is sufficiently adhesive to prevent the ingress of moisture and other liquids but which is elastic enough not to provide any additional shear load which might reduce the strain transmitted to the gauge. Many proprietary materials exist which may be used for this purpose but it is better to employ one which does not involve an abrupt change in temperature in its application—waxes, for example, may be applied to the gauge installation by melting and pouring on to the surface but it has been found that the sudden chilling which occurs if the gauge installation is at room temperature when the wax is applied usually spoils the bond. The difficulty may be reduced by raising the temperature of the gauge installation, but this may not be feasible in heavy engineering applications.

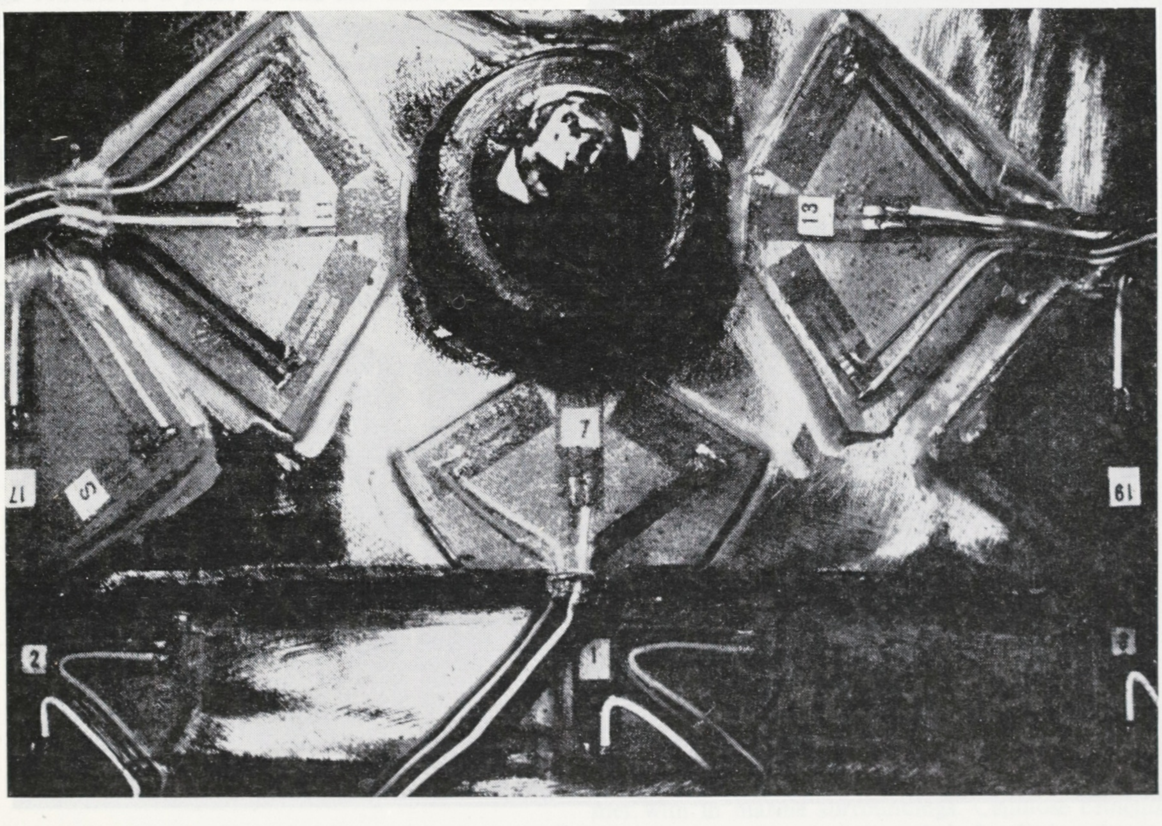


FIG. 8

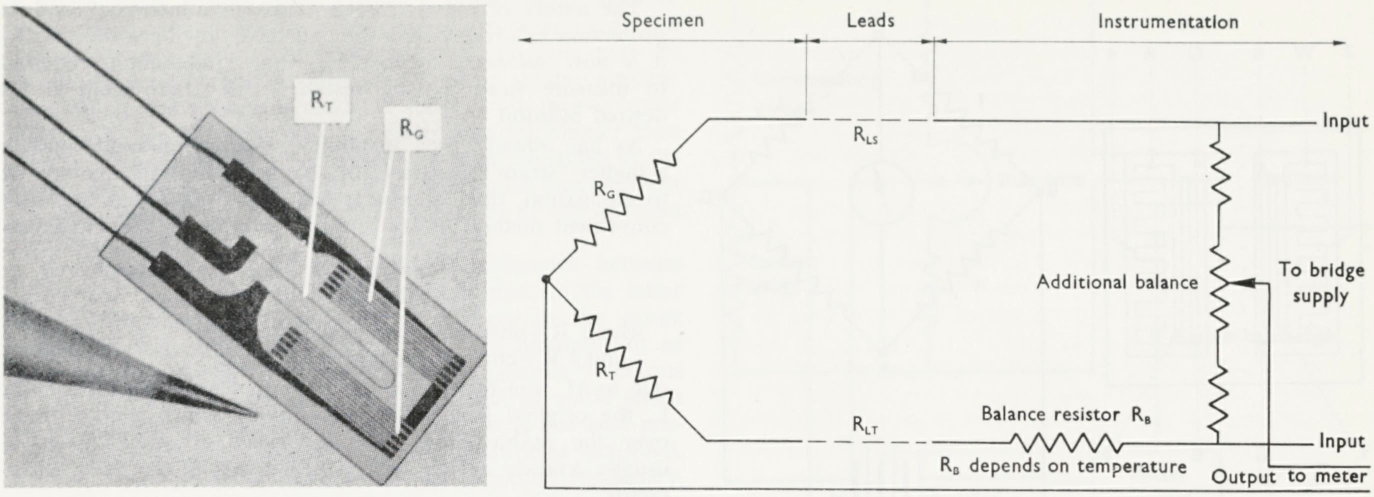


FIG. 9
Self compensating gauge

Therefore a system of encapsulation has been developed which gives in addition, easier handling of the gauges.

The technique is to attach the gauge temporarily to a flat surface by means of double sided adhesive paper tape. Wire leads are soldered on to the gauge terminations, and after this the whole is covered with "Araldite resin 101" plasticised with dibutyl phthalate. After the resin has cured—about 12 hours at 20° C., the completed capsule may be detached from the adhesive tape, the gauge now being in the lower surface of the capsule and level with it. In use the entire capsule is cemented to the test position. If several readings are required at one position, for example when principal stresses are being computed, the necessary gauges are correctly positioned before encapsulation so that they will be retained in their correct relative positions when the capsule is cemented to the test surface. This method of waterproofing is satisfactory for exposure to normal outdoor conditions but where resistance to high hydrostatic pressures is necessary as in hydraulic pressure testing, a coating of thiokol mastic is also applied. This will normally protect a gauge installation up to about 100 atmospheres pressure.

Fig. 6 shows a typical gauge of this type.

The final operation in the installation of a strain gauge, when the cement has cured, is to check the insulation resistance; this should be as high as possible and values of over 20 M Ω are obtainable with proper application.

The Instrumentation used in conjunction with Strain Gauges

In this branch of the subject, it is necessary to sub-divide into two sections as static and dynamic conditions demand very different approaches. It is generally true to say that in static work one is involved with single readings of steady strain from a large number of gauges, while in dynamic measurement simultaneous recording is effected from a relatively small number of gauges; also in the majority of dynamic strain measurements, the slow variation of resistance which may occur in a strain gauge due to temperature change is of little consequence as it is a much slower variation than that due to the true strain.

Static Strain Measurement

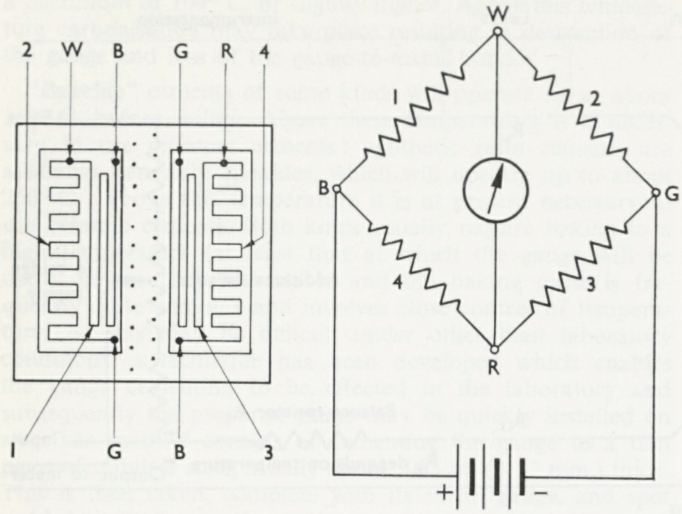
The main problem in static strain measurement may be stated to be that of securing stable, drift-free operation of the gauges.

There are two principal causes of false indication from a strain gauge under "static" conditions, if we exclude humidity effects, which must in all cases be eliminated by suitable waterproofing.

The first is resistance change due directly to temperature rise affecting the resistivity of the wire or foil material. This may be minimised by using gauges made from materials having a low temperature coefficient of resistivity such as Eureka (U.K.) Constantan (Europe) or Advance (U.S.A.) all of which are basically nickel-copper alloys.

The second factor is that temperature change affects most materials dimensionally, and metals, in particular, expand when heated. The strain gauge material, being metal, will also expand when heated, but not necessarily to the same amount as the test piece to which it is attached or the matrix of the gauge plus the cement attaching the gauge. The resultant differential expansion, however, at the gauge material, represents an equivalent strain which also causes a change in gauge resistance. The combination of these two effects produces an "apparent strain" due to temperature change and this must be eliminated, as far as possible, in a gauge installation intended for static strain measurement.

The usual method of achieving this end is to use a "compensating gauge" as shown in Fig. 7. A similar gauge to that used for measurement is attached to a piece of the same material as the test piece and placed as near as possible to the active gauge. Over a period of time the temperature of the active gauge and compensating gauge will tend to equalise and consequently any resistance change due to temperature variation should appear at each gauge equally. Actually, as nothing is perfect, there are disadvantages; for ideal compensation, the compensating gauge should be attached to a piece of material of the same size and shape as that to which the measuring gauge is attached and subject to exactly similar ambient conditions. This is seldom possible and the alternative method usually chosen is to fit the compensating gauge on to a small piece of the same material as the test piece and attaching it near the measuring gauge by a method providing good thermal conductivity without transmitting mechanical strain to it, so that it will follow, as nearly as possible, the temperature change of the test piece. Thus the overall resistance changes due to temperature changes should be nearly equal



FNWFB-50-12 bridge connections

Bridge colour code*

B-Black G-Green R-Red W-White

*Corresponds to BLH Instrument Coding

FIG. 10

Weldable full bridge gauge

for measuring and compensating gauges and when used as adjacent arms in a bridge circuit no unbalance should result from temperature changes.

The system has its disadvantages, but over several days under normal atmospheric conditions in the field, apparent strains may be limited to less than 30–40 by this means. Under test shop or laboratory conditions almost perfect compensation may be obtained, provided large or rapid variations in ambient temperature do not occur.

Fig. 8 shows strain gauges attached to a pressure vessel.

A development of the technique outlined above which may be used in certain cases is that usually known as "self-compensation". If for example, a shaft subject to torsional loading is fitted with gauges located close together, all the gauges will be subjected to nearly the same temperature and when connected in a bridge circuit the unbalance due to temperature change should be insignificant.

It is also possible to manufacture a strain gauge which is, over a certain temperature range, self-compensating. This is done by building into the gauge a temperature sensing "loop" which has a high temperature coefficient of resistivity but by virtue of its low resistance, does not exhibit a change in resistance due to strain which is comparable with that of the strain gauge.

The circuit of such an arrangement is shown in Fig. 9 and this method has advantage at elevated temperatures where conventional methods of compensation may not, for certain reasons, prove effective.

A further refinement is the Baldwin SR4 weldable full bridge (*see* Fig. 10) which has even better self compensation.

The use of such gauges, however, is not normally justified at low ambient temperatures due to their high cost and the satisfactory performance of standard type gauges under the latter conditions.

The nature of the resistance gauge, its attachment, and its protection in adverse environments have now been dealt with; it is now necessary to consider what equipment is required to measure strain having installed the strain gauge in its desired position.

As has already been mentioned, certain materials possess a higher "strain sensitivity" or change in resistance compared to elongation, than others. It has been found that the most convenient method of relating the variables is the expression

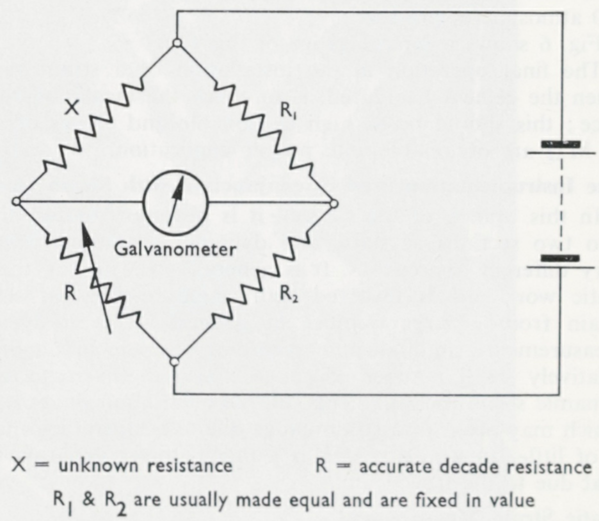
$$\frac{\delta R}{R} \div \frac{\delta L}{L} = S$$

where R = unstrained resistance of gauge,
and δ R = change in resistance,

due to δ L, the increase or decrease in length compared with L, the original length. The quantity S, which is a constant over the usable range of most types of strain gauge is usually known as the "strain sensitivity factor" or "gauge factor".

The quantity $\frac{\delta L}{L}$ is the mechanical strain to which the gauge is subjected, and $\frac{\delta R}{R}$ the corresponding fractional change of resistance. The value of S for the majority of resistance strain gauges is about two, while for semi-conductor gauges the value for types developed up to the present time is about 120 to 140.

As the two types of gauge have such widely different "gauge factors" it is obvious that the appropriate instrumentation required for them will be rather dissimilar. The majority of present strain gauge experience is built up on the resistance strain gauge and as the high cost of semi-conductor gauges will probably preclude their universal use for a number of years yet, especially at higher temperatures, it is proposed to deal with instrumentation devoted to the resistance type of gauge; this may be applied, with certain modifications to the semi-conductor gauge, but as it has so much higher a gauge factor the modifications will in fact always be simplifications.



X = unknown resistance R = accurate decade resistance

R_1 & R_2 are usually made equal and are fixed in value

FIG. 11

Wheatstone bridge

Static strain measurement covers a rather broad field of work ; it may be that only a few gauges are being used on a laboratory application, or perhaps some hundreds may be employed on a structure, pressure vessel or similar application. While the apparatus used for static strain measurement may as a rule be electrically (or electronically) simpler than that used for dynamic measurements, it must be stated at the outset that problems associated with static strain measurement generally render this a more difficult branch of the art than dynamic measurement.

As already mentioned, the principal difference between static and dynamic strain measurement is that, in the latter strains are varying so rapidly that changes in the gauge resistance due to temperature variation are unimportant as they are too slow to be confused with the genuine variations of strain. Returning therefore to the subject of static strain measurement, the main point to be ensured is the best possible temperature compensation. With this point achieved, although this is one of the least simple conditions to satisfy, the measurement of static strains is a matter of accurate measurement of small changes of resistance.

The conventional Wheatstone's bridge circuit (Fig. 11) immediately springs to mind, but this is generally a rather clumsy device to use with strain gauges, at least in its conventional form.

A variant is the slide-wire bridge shown in Fig. 12, in which the unbalance produced by a resistance change at the gauge is nullified by a movement of the slide-wire to restore balance.

In Fig. 12, with the ratio arms X ohms and the slide-wire terminal resistance Z ohms, if the change in slide-wire setting from initial balance x ohms, this will indicate a change of resistance at the strain gauge.

$$\frac{\delta R}{R} = \frac{x}{2X+Z}.$$

Thus if S is the "gauge factor" of the strain gauge in use,

the strain $\dfrac{\delta L}{L} = \dfrac{x}{S(2X+Z)}.$

While Fig. 12 shows this circuit arranged as a D.C. bridge its more usual practical form in present-day work is the A.C. bridge of Fig. 13. The sensitive galvanometer may then be replaced by a robust milliameter which is fed with the amplified bridge output via a valve or transistor amplifying circuit. The amplifier can be sharply tuned so that it only responds to signals at the bridge excitation frequency and not to signals

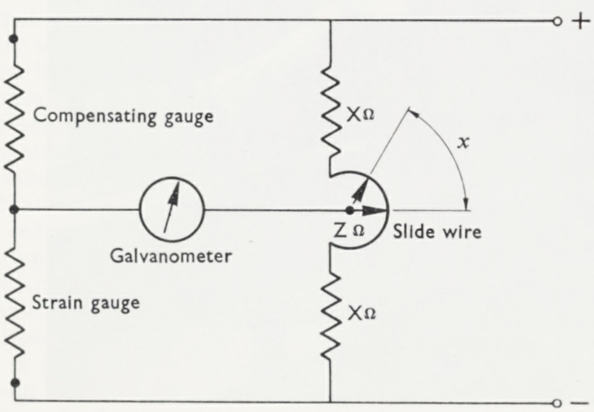


FIG. 12
Slide wire bridge

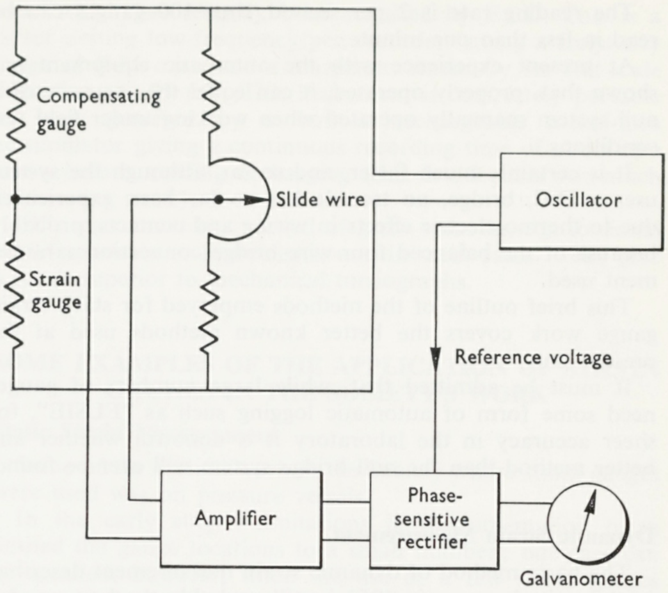


FIG. 13
Schematic circuit of Baldwin A.C. bridge

such as, e.g. interference from power supply fields or possible thermal potentials in the gauge circuit.

As the bridge is adjusted to the null point (i.e. zero output) for measurements, the sensitvity of the amplifier is immaterial provided that it is sufficient to indicate the small unbalances likely to be experienced. Its sensitivity does not contribute otherwise to the accuracy of measurement which is dependent upon the accuracy and stability of only the slide-wire and ratio arms, X, X and Z.

A picture of a typical strain bridge of this type is shown in Fig. 14.

Such instruments have been used for measuring from up to several hundred gauges using selector switching but when the number of gauges exceeds about 100 it becomes a tedious method. When large numbers of gauges are being employed for static strain measurements, the modern tendency is to use an automatic data logging system and with this end in view the Society's Engineering Investigation Department have recently developed in conjunction with Solartron Ltd. the device shown in Fig. 15 and known as "ELSIE" (Electronic Strain Indicating Equipment).

This apparatus consists, very basically, of a large number of bridge circuits (up to 800 may be used) in each of which two arms are formed of a strain gauge and temperature compensator and the other two by ratio arms ; a balancing adjustment is provided in the ratio arm circuit.

An automatic switching system connects in sequence the bridge supply and a digital voltmeter to each bridge ; the bridges are initially balanced when the strain gauges are in the unstrained state at the commencement of a test. As the gauges are strained, a voltage appears across the "detector" points of the bridge (*see* Fig. 16) and this is read by the digital voltmeter. By choosing the bridge voltage correctly, the reading of the digital voltmeter can be arranged to indicate strain directly.

Automatic reading and printing facilities up to 100 positions are provided, manual change to the next 100 gauges being necessary at present.

The reading rate is 2 per second, thus 100 gauges can be read in less than one minute.

At present experience with the automatic equipment has shown that, properly operated, it can equal the accuracy of a null system manually operated when working under field test conditions.

It is certainly much faster, and so far, although the system uses a D.C. bridge, no trouble has so far been experienced due to thermo-electric effects in wiring and contacts, probably because of the balanced four-wire bridge connection arrangement used.

This brief outline of the methods employed for static strain gauge work covers the better known methods used at the present time.

It must be admitted that, while large numbers of gauges need some form of automatic logging such as "ELSIE", for sheer accuracy in the laboratory it is doubtful whether any better method than the null bridge system will ever be found.

Dynamic Strain Measurement

The basic method of dynamic strain measurement described in the writer's paper in 1953 is still probably the best one for general use.

In detail, however, many improvements have been possible in the last decade.

The ultra-violet type of recorder, in which a beam of U.V. light from a mercury vapour lamp is reflected by a mirror-type galvanometer on to self-developing photographic paper, is now used for most purposes and the type used by E.I.D.

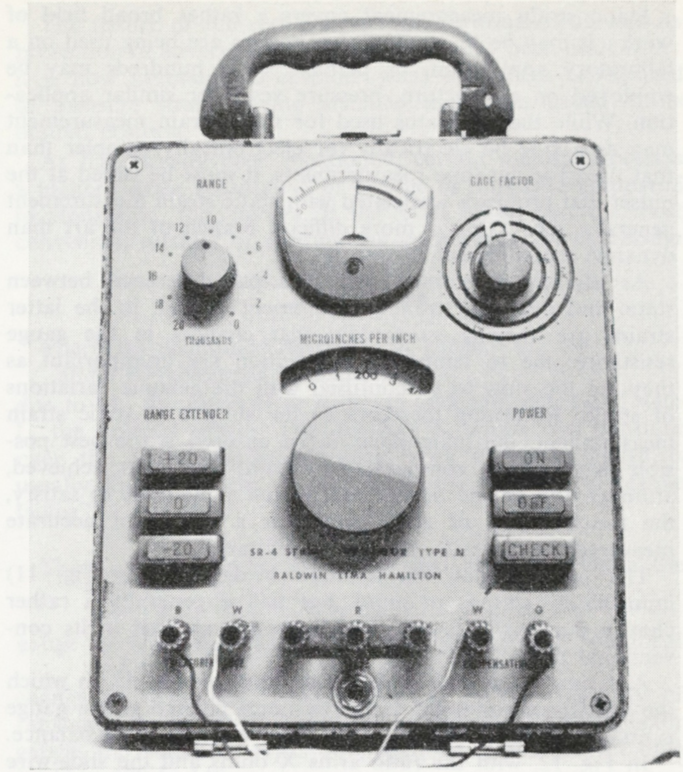


FIG. 14

FIG. 15
"ELSIE" (Electronic Strain Indicating Equipment)

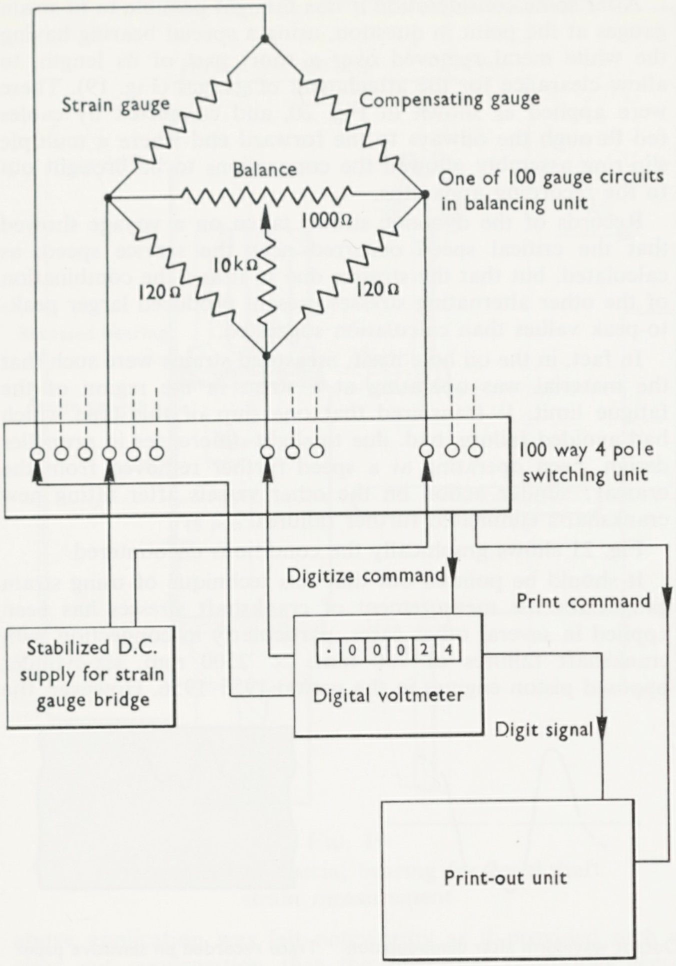


FIG. 16

Schematic diagram of "ELSIE"

has a frequency range extendible up to about 3000 cycles per second if suitable galvanometers are used. The older type of apparatus used a pen recorder which had a frequency limit of about 100 c/s. Also, the use of transistors for the amplifier circuits has reduced size, weight and power supply requirements and presently available apparatus recording up to 12 channels of simultaneous strain phenomena is only as bulky as that used previously for four channels, while possessing higher sensitivity.

The higher sensitivity obtainable with the ultra-violet type of recorder has enabled satisfactory operation to be attained with only three volts bridge supply, leading to the use of 120 ohms strain gauges, which, due to their low impedance do not necessitate the employment of shielded cables in normal applications. Due to the low bridge power, thermal drift is also reduced.

A typical arrangement of 12 channel recorder and amplifier together with their associated bridge balance arrangements is shown in Fig. 17. The schematic diagram of Fig. 18 shows the basic circuit used for one channel.

It is possible to operate strain gauges into this equipment via slip rings, enabling measurements of vibration stresses in shafts and other rotating members to be measured.

Mention will also be made, for the sake of completeness, of another instrument suitable for dynamic strain measurement which has recently been acquired by E.I.D. This is a direct writing low-frequency pen recorder having a frequency range of 0–85 c.p.s. and a maximum sensitivity for full scale deflection of ·25 mV. This instrument is completely portable as it is powered by a built-in rechargeable nickel-iron accumulator giving a continuous recording time of about two hours before needing recharging (this is adequate for much field testing work).

This instrument has proved of great value in, e.g. torsional vibration strain measurements, and in most cases it can give a result superior to mechanical torsiographs.

SOME EXAMPLES OF THE APPLICATION OF STRAIN GAUGES IN THE SOCIETY'S WORK

Static Strain Measurement

Some of the earliest work in the Society where strain gauges were used was on pressure vessels.

In the early stages, limitations in instrumentation often limited the gauge locations to a small number; one case can be recalled, however, when the use of a few strain gauges at a welded transition from large to small diameter sections in a refinery reactor vessel gave considerable confidence to the designers by confirming their stress estimates under pneumatic test conditions.

Indeed, pressure vessel testing now occupies a very important position in the work of the Engineering Investigation Department and the strain gauge installation and protection techniques previously described have been developed originally for use in this field of work.

Commencing with Calder Hall, many of the reactor pressure vessels for nuclear power stations in this country and elsewhere were subjected to strain measurement under pneumatic test, which form of testing is necessitated in large vessels where the weight of such a volume of water on the foundations precludes the possibility of hydraulic testing. The hazards involved with pneumatic testing indicate the desirability of investigating the strains at critical points in the vessel and the Society's Engineering Investigation Department have in most cases applied several hundred gauges at strategic points on each of the vessels tested. The data obtained have

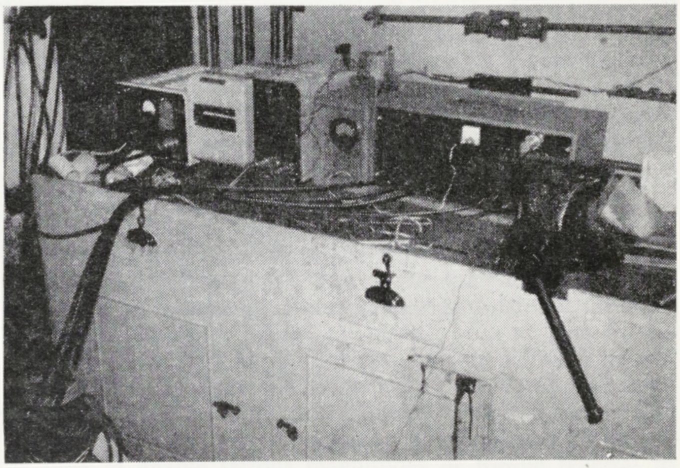


FIG. 17

12-channel strain recording equipment

subsequently enabled advances to be made in design of later pressure vessels.

A part of the gauge installation on a pressure vessel is shown in Fig. 8 and the equipment for reading the strain in Fig. 15.

Structural testing using strain gauges on a large scale commenced in the early 1950's when the Society was called upon to strain gauge several double bottom sections for B.S.R.A. Since this time several other applications in ships' structures have been found, especially recently in connection with fork-lift truck loads on deck sections.

Dynamic Testing

(1) Measurement of Crankshaft Strains

A type of fast cargo liner having a ten cylinder two-stroke-cycle oil engine exhibited cases of crankshaft failure at the same position involving cracking in the region of the oil hole in the centre journal.

This point was very close to the position of the crankshaft node of the two-node mode of torsional vibration of the machinery and the cracking exhibited typical 45 degree orientation characteristic of torsional fatigue failure. Examination of the torsional vibration calculations revealed that the eighth order two-node critical occurred fairly close to the service speed but that the stresses predicted were quite small.

After some consideration it was thought possible to fit strain gauges at the point in question, using a special bearing having the white metal removed over a short part of its length, to allow clearance for the attachment of gauges (Fig. 19). These were applied as shown in Fig. 20, and connected by cables fed through the oilways to the forward end where a multiple slip-ring assembly allowed the connections to be brought out to the recording apparatus.

Records of the dynamic strains taken on a voyage showed that the critical speed occurred near the service speed, as calculated, but that the stresses due to it and the combination of the other alternating stresses present produced larger peak-to-peak values than calculation suggested.

In fact, in the oil hole itself, measured strains were such that the material was operating at a stress in the region of the fatigue limit. It transpired that one ship of this type which had avoided failure, had, due to slight differences in propeller design, been operating at a speed further removed from the critical; similar action on the other vessels after fitting new crankshafts eliminated further failures.

Fig. 21 shows graphically the conditions encountered.

It should be pointed out that this technique of using strain gauges for the measurement of crankshaft stresses has been applied in several other cases, particularly in connection with crankshaft failures of 750 mm. × 2500 mm. six-cylinder opposed piston engines in the period 1955–1956. However, the

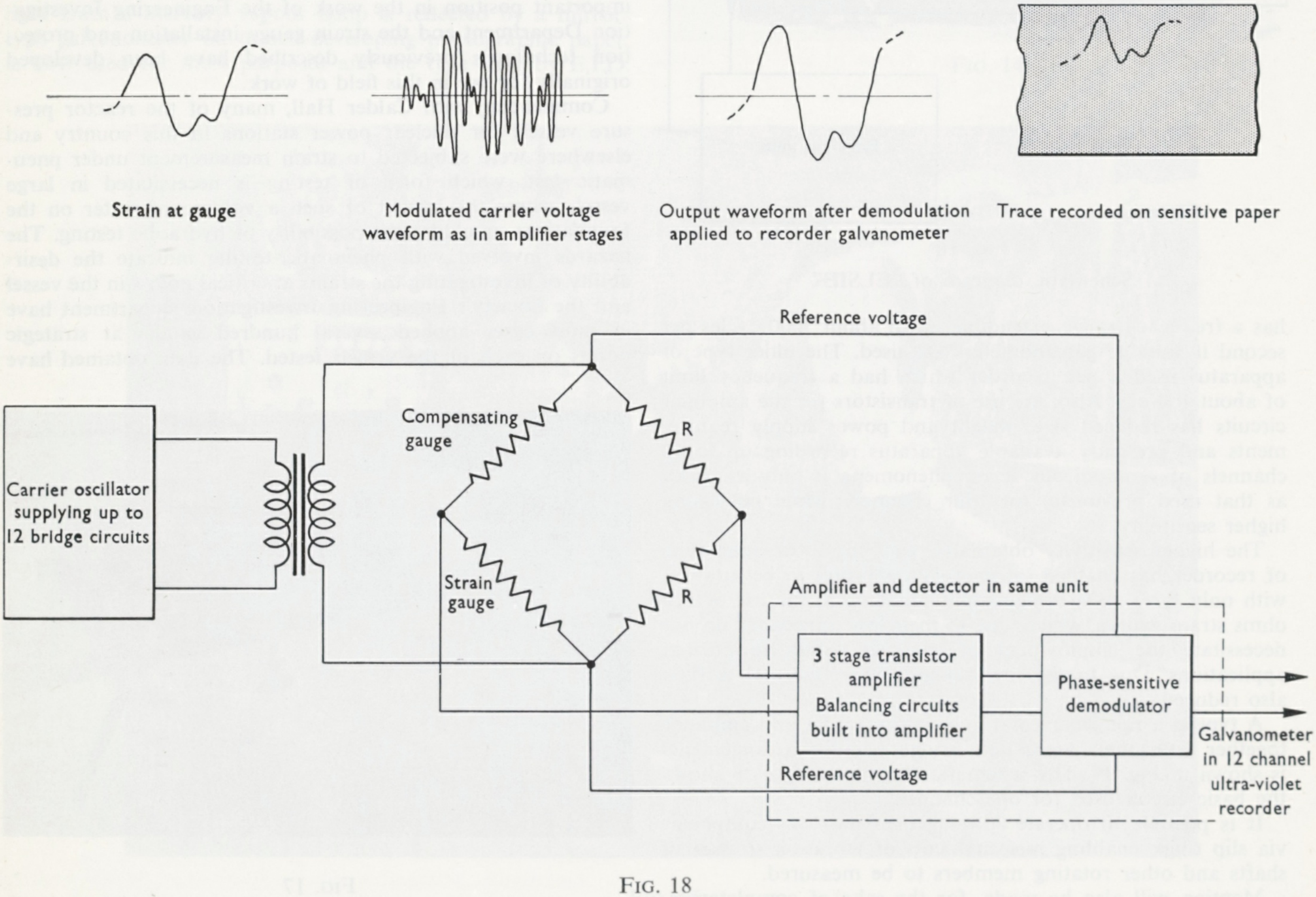


Fig. 18

Schematic circuit of one channel of 12-channel strain recorder

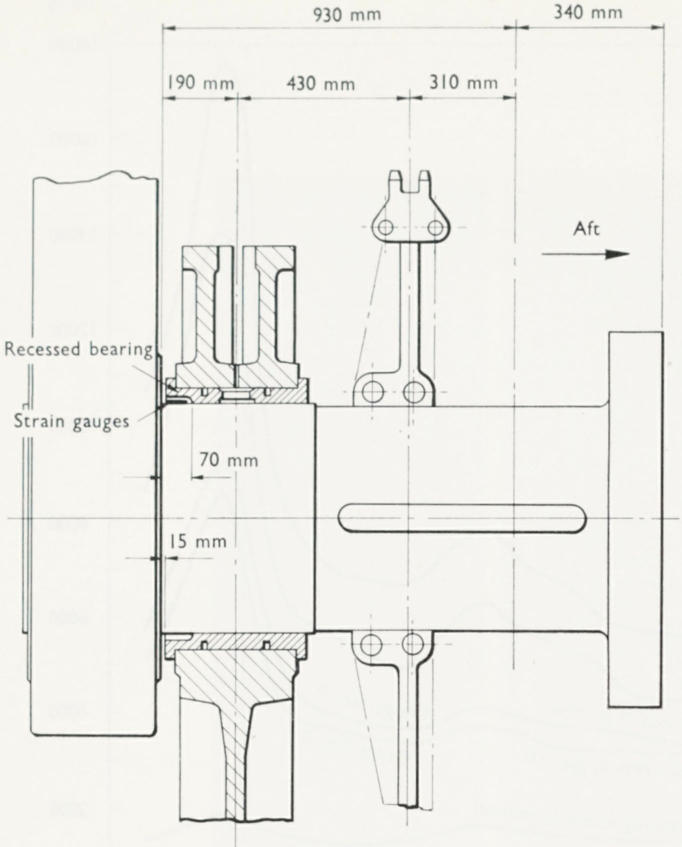


FIG. 19
Arrangement of special bearing for crankshaft strain measurement

above application was felt noteworthy as it provided such a clear cut confirmation that the operation conditions were contrary to theoretical indications.

Recommendations were made to operate at a speed further removed from the critical r.p.m. and attention was drawn to the rather rough internal finish of the oil holes; this was improved by reaming the outer region of the holes *in situ*.

(2) FAILURE OF A TORSIONAL SPRING COUPLING

A spring coupling used to provide torsional flexibility between a high speed vee-diesel engine and a D.C. generator had suffered frequent and rapid fatigue failure of the spring elements.

One of the unknown conditions of operation was the stress during starting and run-up, which could not be readily deduced from ordinary torsiograph readings, as the rapid acceleration under these conditions rendered such an instrument unreliable. Therefore strain gauges were attached to six of the spring elements in a 12-element coupling.

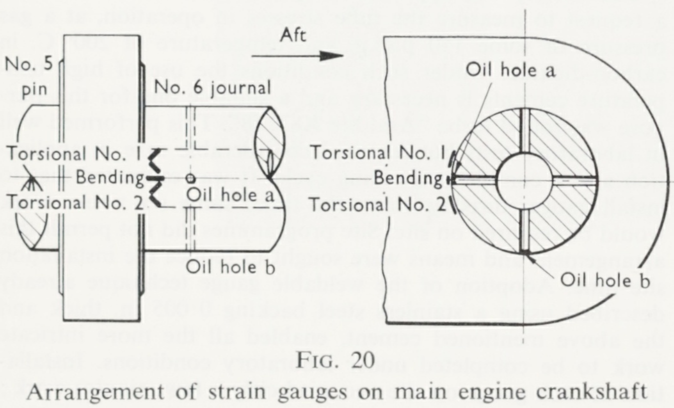


FIG. 20
Arrangement of strain gauges on main engine crankshaft

Connections were brought out from the gauges through holes in the case of the coupling, the holes being sealed with "Araldite" cement to prevent leakage of the crankcase oil which was allowed to fill the coupling for lubrication purposes via leakage from the aftermost crankshaft main bearing.

Fig. 22 shows the positions of the strain gauges on the coupling springs.

A typical record taken during start-up (Fig. 23) shows that an excessive alternating stress was applied to the springs during running up, although at service speed the stresses were well within the capability of the material.

As the only time at which the springs were over-stressed was during this transient condition, stops were fitted to limit the excursion of the springs from the mean position during start-up, whereby the flexibility under normal operating conditions was unaffected, as the stops only came into contact with the springs under the transient condition. Unfortunately time did not permit a repetition of the readings with the stops in place but no further casualties have been reported.

(3) TORSIONAL VIBRATION IN SYNCHRONOUS ELECTRIC DRIVES

During the run-up of a synchronous, salient-pole A.C. motor, alternating torques are generated, in addition to the mean accelerating and driving torque, which vary in frequency at 2 $\times$ the slip frequency, i.e. from 100 c/s to zero in a 50 c/s machine.

(This phenomenon is inherent in the torque equation of a salient pole machine.)

At a steel works in S. Europe, a blower installation for a blast furnace was set up which consisted of two identical sets of centrifugal compressors, gearboxes and driving motors, these latter being synchronous machines having six salient poles.

During commissioning, it proved impossible to start up these sets, due to a violent torsional vibration critical speed at about 50 per cent of the full motor speed. A check on the natural frequency of the system showed that the one-node critical frequency was approximately 50 c/s.

As the exciting frequency from the motor is equal to twice the slip frequency, this, in fact, gave a critical speed at about 50 per cent of the service speed of 1000 r.p.m.

Fitting a Holset type flexible coupling enabled the natural frequency of the system to be reduced to aboupt 7 c/s, which meant that the critical speed should now occur at approximately 92/93 per cent of the synchronous speed.

As the gearboxes had been damaged in initial attempts to start the machines, strain gauges were fitted to the cages of the epicyclic system (Fig. 24) to enable the relative torque to be measured on start-up. Records taken showed that the critical speed did occur where calculated; due in part to the damping introduced by the flexible coupling and also in part to the extremely rapid rate of change of excitation frequency which occurred with the new position of the critical speed, torque fluctuation did not occur over a long enough period of time to enable serious magnitudes of torque reversal to develop.

Fig. 25 shows graphically the torque conditions during start-up.

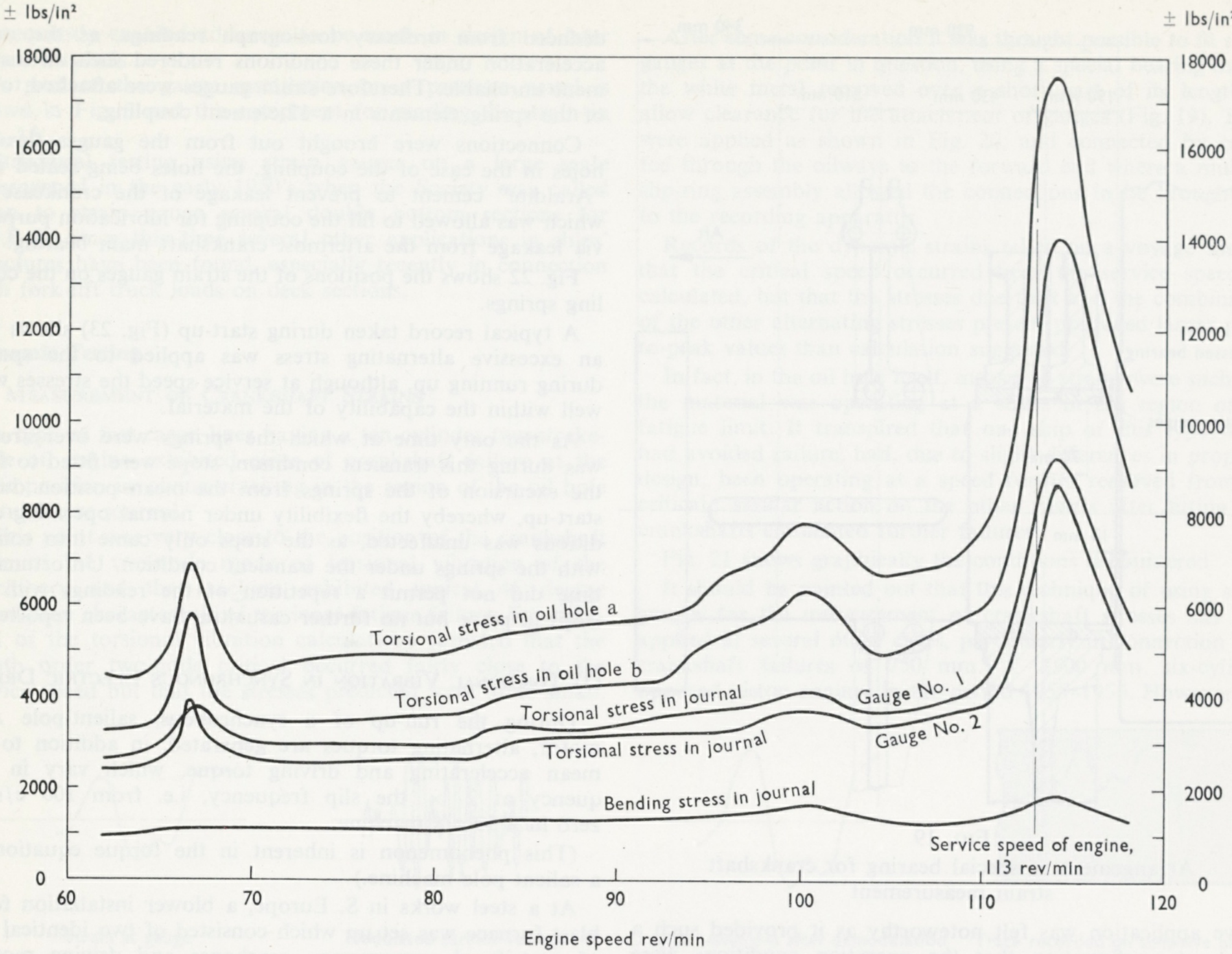


FIG. 21

Crankshaft stresses recorded in main engine journal No. 6

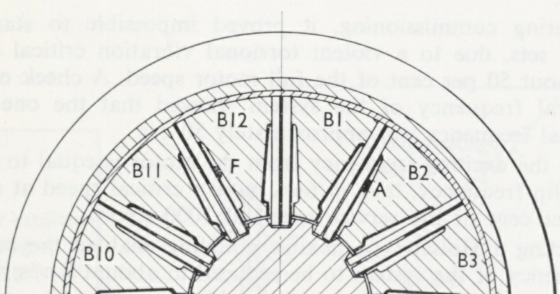


FIG. 22

Simplified view of torsional spring coupling showing strain gauge positions A to F on radial spring elements

(4) VIBRATION OF HEAT EXCHANGER TUBES

During preliminary testing, the main heat exchangers of a large nuclear power plant suffered several tube failures. The failures were localised and consistent with fatigue of the material at the junction of the tube, with a welded hanger; the only exciting force which appeared feasible was that due to vortex-shedding from the tubes in the gas-flow.

The Society was approached by the main contractors with a request to measure the tube stresses in operation, at a gas pressure of some 150 p.s.i.g. and temperature of 200° C. in carbon-dioxide. Under such conditions the use of high temperature cements is necessary and a suitable one for this purpose was found to be "Araldite X83/78". This performed well in laboratory tests but required considerable care in application and a complicated curing cycle. It was estimated that to install gauges directly on to the tubes some six weeks work would be required on site. Site programmes did not permit this arrangement and means were sought to reduce the installation site time. Adoption of the weldable gauge technique already described using a stainless steel backing 0·005 in. thick and the above mentioned cement, enabled all the more intricate work to be completed under laboratory conditions. Installation of each gauge on site occupied only a few minutes work;

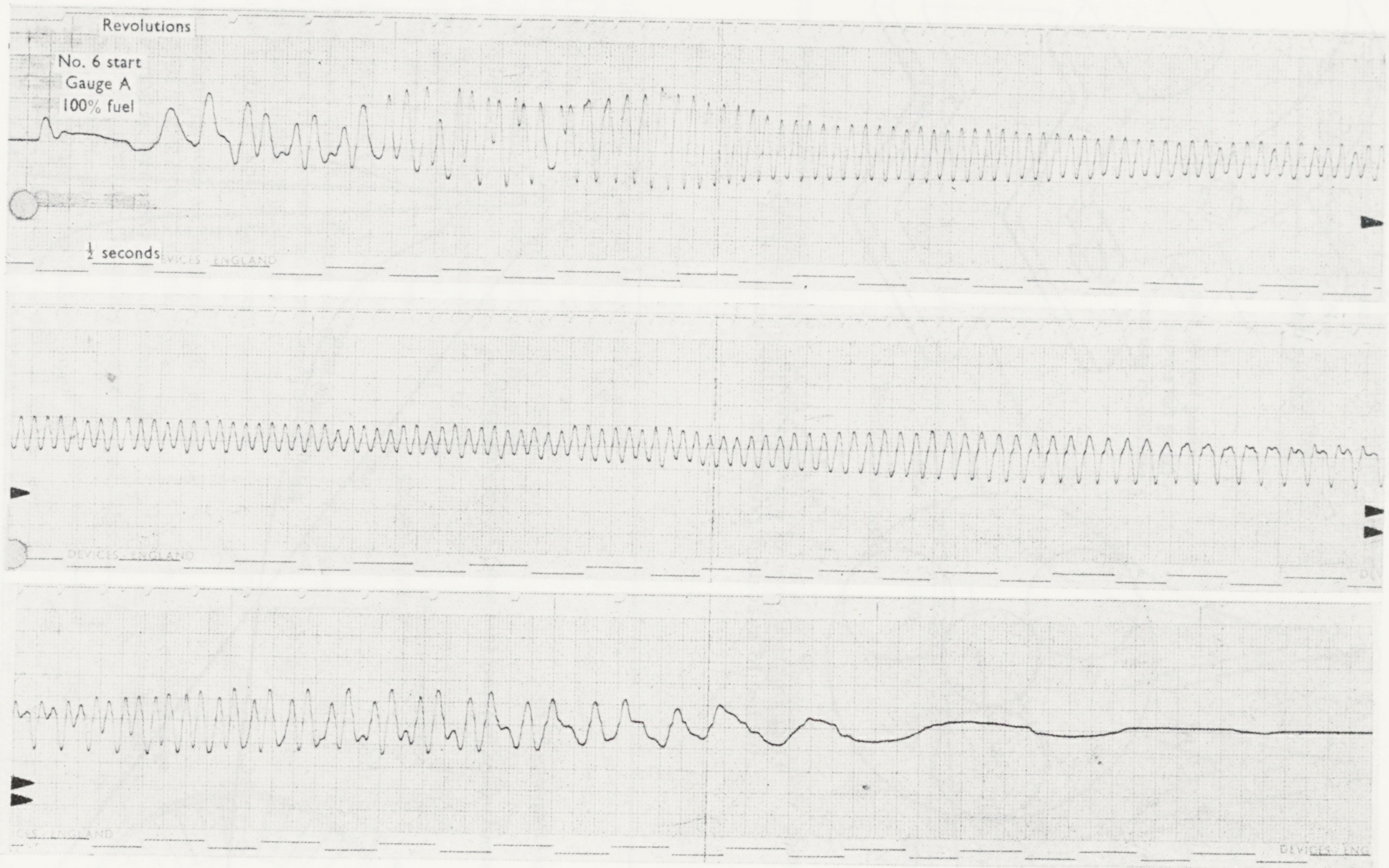


FIG. 23

A record from one of the strain gauge positions (A) shown in Fig. 22. The large dynamic stresses just after starting were causing premature failure of the coupling springs

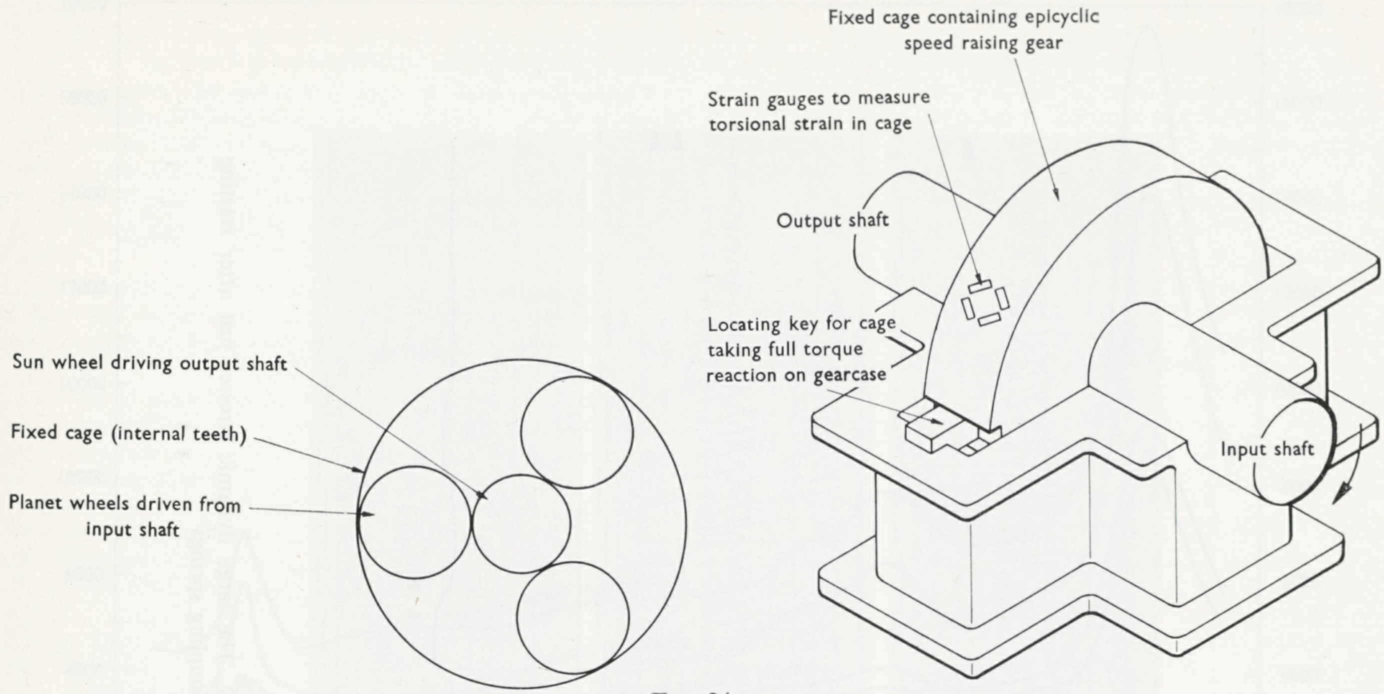


FIG. 24

Diagrammatic view of gearing showing strain gauges attached to indicate in proportion to torque in system

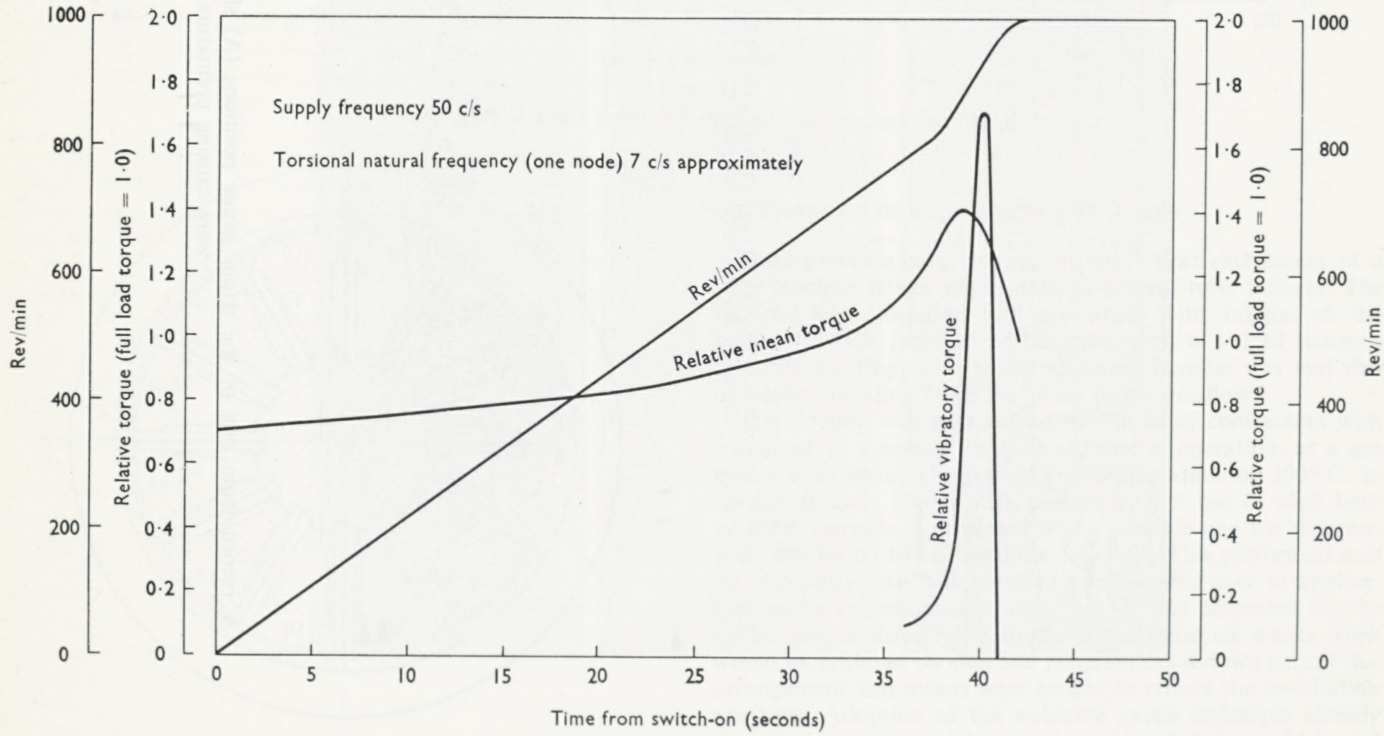


FIG. 25

Typical start and run up conditions for direct switch-on

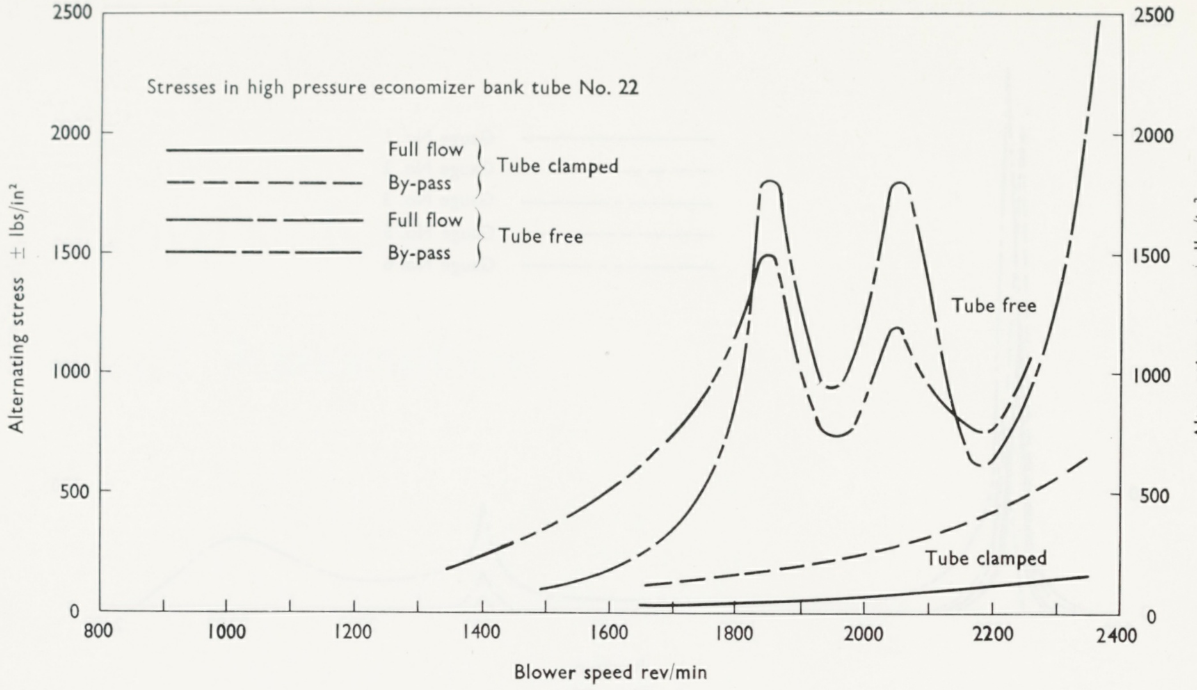


FIG. 26

Curve showing stress levels under working conditions in economizer tube of Nuclear power plant heat exchanger. Note reduction in dynamic stress due to clamping of tube

site work was further minimized by using solderless connections between the gauges and wiring. Wiring having P.T.F.E. insulation was used, and the total installation time for about 150 gauges was under ten days. Subsequent to the initial failures, the majority of the tubes had been stiffened by additional clamping; the difference between stresses in typical free and clamped tubes is adequately demonstrated by Fig. 26.

The heat exchangers have operated without incident since the clamping of all the previously "free" tubes.

(5) FATIGUE FAILURE OF BLOWER GUIDE VANES

Difficulty was experienced due to fatigue failures in the guide (stator) vanes of the CO_2 blowers of a nuclear power plant. In the confined space inside the blowers, fitting strain gauges by conventional methods would have been difficult, if not impossible.

The weldable gauge technique was again employed, gauges being attached as shown in Fig. 27 to 12 of the vanes.

Connections from the gauges were brought out through a swan-neck tube attached to the pressure balancing connection and sealed with "Araldite" cement.

Records of the dynamic strain were taken over the operating range and at the 1,000 r.p.m. region a critical speed due to blade passage excitation between rotor and stator was located (Fig. 28). It was determined that prolonged running at this speed had taken place during start-up and since this region of the speed range was required for this purpose a modified blade design was fitted which had a higher natural frequency and thereby removed the resonant condition from the operating speed range.

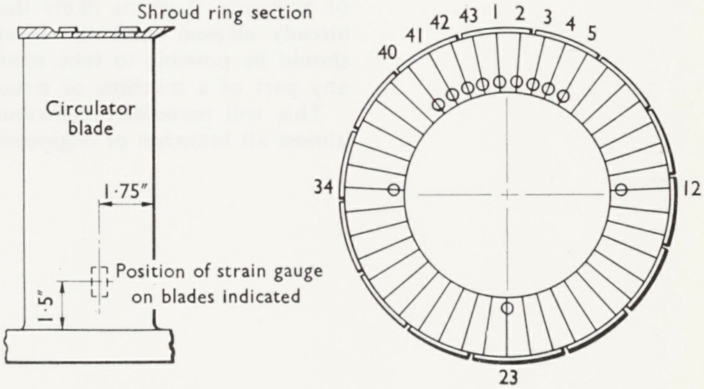


FIG. 27

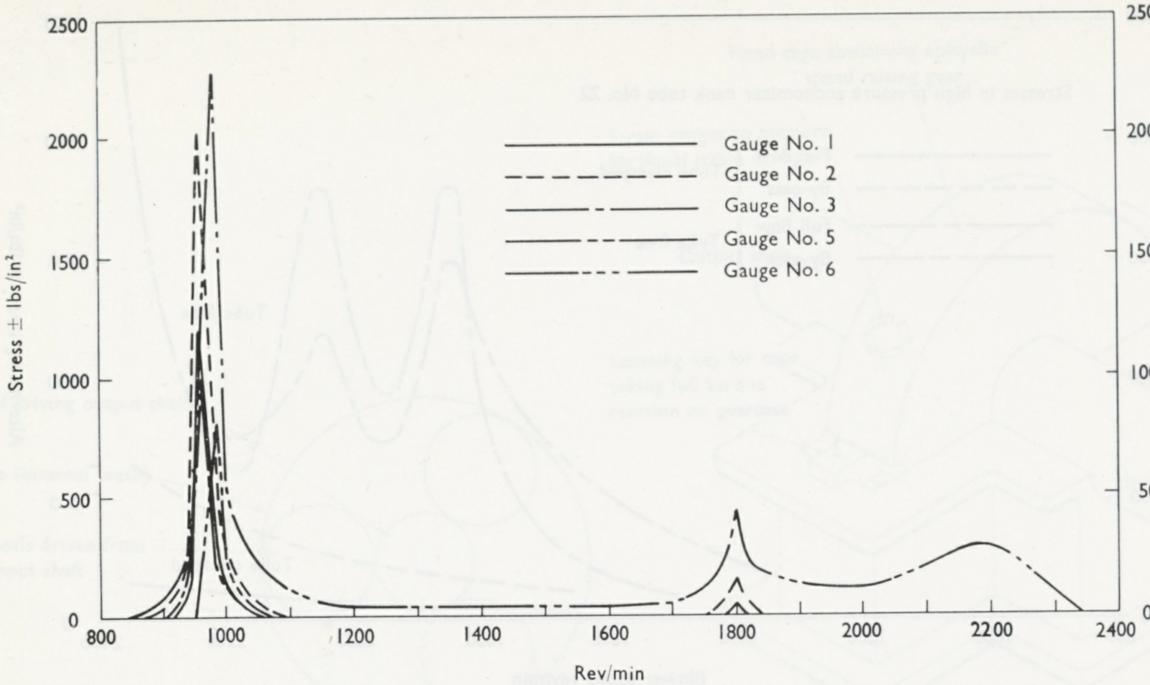


FIG. 28

Stresses in CO_2 blower guide vanes

SUMMARY

In concluding, the Author of this short and by no means complete exposition of present-day strain gauge techniques would point out that the present state of the art of strain measurement has been reached at about 25 years from the date of invention of the strain gauge.

Progress in the development and use of the device has been rapid, due in part to the equally rapid development of electronics in the same period.

If one can look to the future and predict with any degree of accuracy, it seems likely that in a shorter time than has already elapsed since the inception of the strain gauge, it should be possible to take strain measurements from almost any part of a machine or structure.

This will materially contribute to safety and economy in almost all branches of engineering.

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GLASS REINFORCED PLASTIC BOAT BUILDING — PARTS III AND IV

by

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GLASS REINFORCED PLASTIC BOAT BUILDING—PARTS III AND IV

By W. L. Hobbs and A. McInnes

PART 3—CONSTRUCTION

By W. L. Hobbs

PART 4—CONSTRUCTION DESIGN

By A. McInnes

PART 3—CONSTRUCTION

By W. L. Hobbs

3.1 INTRODUCTION

The production in this country of reinforced plastic marine craft of any real size was slow to start, mainly due to the fact that a number of firms had been producing dinghies and small launches without proper facilities, and by producing craft that did not stand up to all the fictitious claims made for them, engendered doubts in the minds of the public regarding the strength and durability of craft made of this material.

One British firm was engaged in the production of marine craft in G.R.P. as far back as 1954, and produced shortly after that date, the largest one-piece moulding of this material in the world. This firm had the foresight to call in the services of the Society at a very early stage of its life and the result has been the restoration of confidence in the material, based upon satisfactory service given by craft produced by this firm under varying climatic conditions both in the commercial world and private pleasure service, including service in the home of G.R.P. pleasure craft, the U.S.A.

First-hand experience gained from craft produced by this and several other firms was of much assistance to the Society in the preparation of the Provisional Rules for the Construction of Reinforced Plastic Yachts.

Since these Rules were published the number of firms producing sizeable craft of this material has increased and most of them were quick to realise the benefits derived from calling in the Society's research and survey facilities. It seems rational to assume the Society will be called upon to deal with an increasing volume of this type of construction and it is hoped this paper will prove helpful to colleagues who are called upon to survey reinforced plastic marine craft during construction. With this end in view, the paper has been confined as far as possible to the practical side of construction.

There is little doubt that research will provide us with new resins and possibly new reinforcement materials and moulding techniques may be developed to render this paper obsolete in a very short time, but although new resins may come fairly

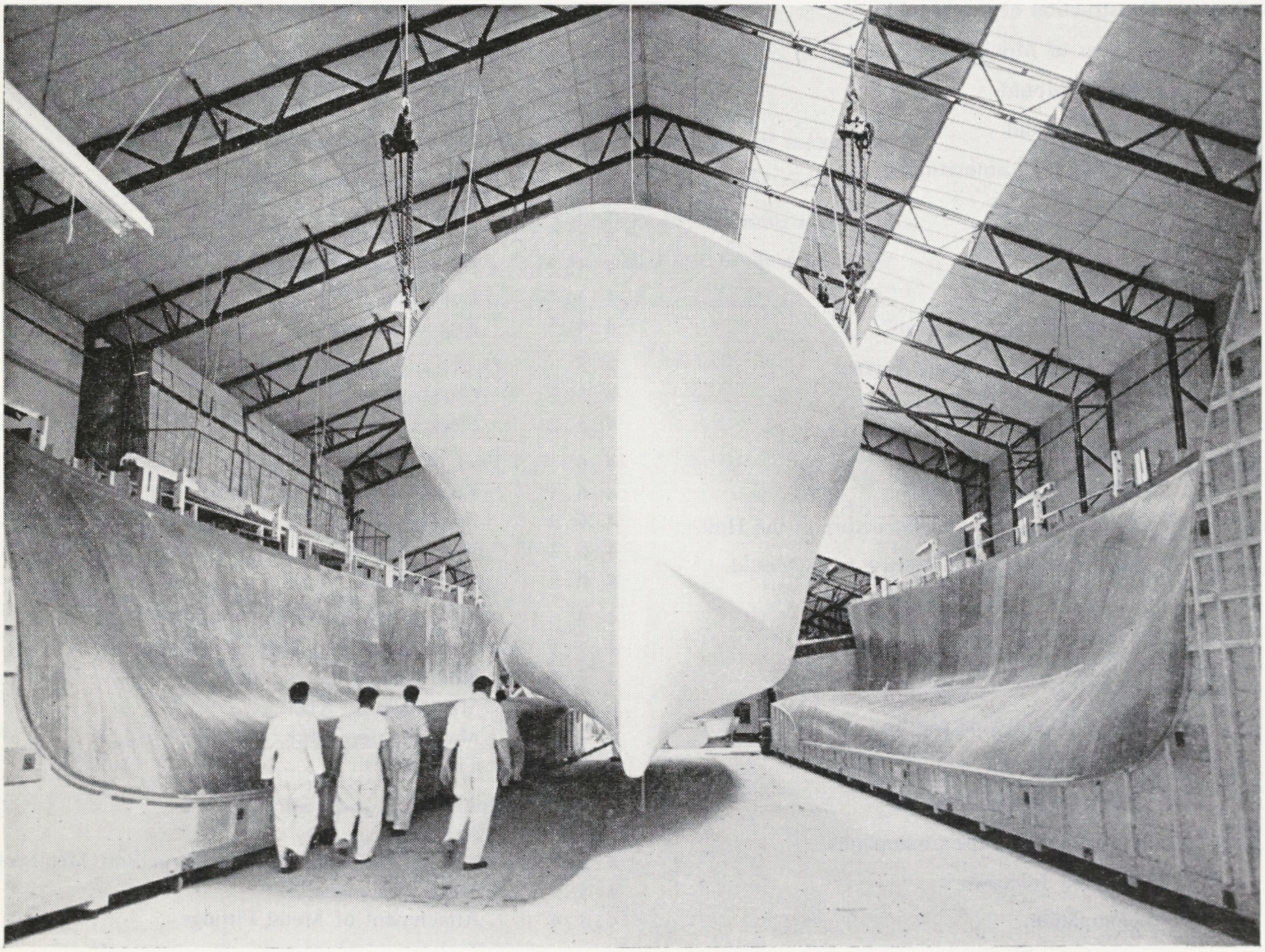


Fig. 1
Hull of a 67 ft. motor yacht free from the mould and suspended from the moulding shop roof girders.

quickly, with the increase in size of craft produced, reinforcing materials already in existence, e.g. steel and aluminium alloy will be incorporated and we may well end up with "Composite" reinforced plastic construction which will embody the well known strength properties of these materials and eliminate their also well known corrosion and erosion problems.

It is right we should all think of things to come and advance with the times, but until the time comes, this paper may serve a useful purpose.

Assuming proper workshop conditions are available, the construction of reinforced plastic craft can be broken down into three main stages:—

(1) Construction of the mould.
(2) Laying-up or moulding.
(3) Fitting-out and completion.

3.2 CONSTRUCTION OF THE MOULD

3.2.1—Types of Mould

The mould for a yacht or boat hull is normally of the female type, for as only the surface of the moulding in contact with the mould presents a smooth finish, it is naturally desirable that this should be the outside surface of the hull.

Moulds can be of wood or plastics and if a large number of mouldings are to be taken from one mould, then the plastics mould, being more durable, wins every time. For very large craft, say 50 ft. or over, this may be ruled out on the score of initial cost, but a well constructed wood mould can be used for up to 12 or 15 mouldings before it needs refurbishing.

The moulds are made with a joint at the centreline in the form of two flanges bolted together, and if the stern has a transom ending it is usual to make a joint here also, to facilitate removal of the moulding. As it is essential that the two halves of the mould and the transom joint should be capable of being parted and re-made to register correctly several times, it is advisable to bush at least some of the securing bolt holes to avoid bad register through wear.

On large moulds, the outer framing is taken down to a level base and fitted with suitably spaced castors so that the two halves can be pulled apart whilst the cured laminate is suspended on overhead gantries and then lowered into the fitting-out cradle. As considerable static can sometimes generate during the hand lay-up process, it is advisable to have metal castors. If rubber tyred castors are used, chains or wires should be attached to the mould touching the moulding shop floor to provide leak channels for the static electricity.

3.2.2—Wood Mould

Good service can be obtained from a double skinned wood mould constructed upon substantial outside frames carrying closely spaced ribbands upon which the two skins of planking are laid up. Very accurate lofting is essential, the material for the outer framing and the skins should be kiln dried to 10 per cent moisture content and the mould constructed in workshop conditions as near as possible to those maintained in the moulding shop.

The outer skin is nailed or stapled and glued to the stringers, the inner surface of this skin is then sanded as smooth as possible and the inner skin then laid upon it, being glued and staple fastened also. The two skins are usually laid diagonally, the grain of the inner skin being at right angles to that of the outer skin. When both skins are laid and the glue has cured, the staples are removed from the inner skin, the surface suitably primed after sanding, staple holes stopped in, and the surface then brought up to a high standard of finish.

This latter is a tedious job, calling for many coats and plenty of rubbing down between coats with wet and dry abrasive, but as the finish of the contacting surface of the mould governs the standard of finish of the moulded article, it is obvious that only the best obtainable finish is good enough. Furane resin has been found excellent for this purpose, for it not only stands up well to the heat generated during laminating, but it requires very little in way of a release agent. One firm even coats its glass fibre moulds with this material for the latter reason.

3.2.3—Glass Mould

Where it is decided that it is economic to use glass fibre moulds, then a plug is made, usually of wood, but sometimes for very small craft, of plaster, the surface brought to the same high standard of finish as for a wood mould, and after applying a parting agent, a gel coat is laid on and when gelled, a second gel laid on, backed with a surfacing tissue, after which the required thickness of the mould is completed with chopped strand mat and resin. A suitable framework is bonded to the outside, and this can be of wood bonded to the glass mould with flanges of chopped strand mat, and castors fitted as necessary. The joining flanges are usually of glass fibre of greater thickness than the shell thickness of the mould, and these are bolted together as in the case of wood moulds.

If wood is used for the plug, the construction can be of the strip planked type on wood frames, or double skin glued construction. It is necessary to take the same precautions in kiln drying the timber for the construction of the wood plug as for wood moulds.

3.2.4—Access for Laminating

Moulds for smaller craft are sometimes made capable of being rotated 90 degrees each side of the vertical so that laminating can be carried out without operatives entering the mould.

If the mould is too large for the laying-up to be carried out from outside of it, then a system of staging must be installed upon which the laminators can work comfortably inside the mould, and this should be done before the mould is finally cleaned and transferred to the moulding shop.

Figure (2) illustrates a typical form of staging.

In addition, it is of course necessary to have staging and steps erected all round the outside of the mould in order that the laminators can obtain easy access to the inside of the mould and so that the materials for laminating can be safely handed to them.

3.3 LAMINATING OR "LAYING-UP"

3.3.1—Preparation of Mould Surface

When the manufacture and finishing of the mould is completed, it is transferred to the moulding shop where it is coated with a parting agent to ensure trouble-free separation from the moulded laminate.

Parting agents can be hard wax followed by a P.V.A. coating, or one of the proprietary preparations sold for the purpose.

If the mould has been finished with furane resin, then a light rubbing with "Cobra" wax is all that is required.

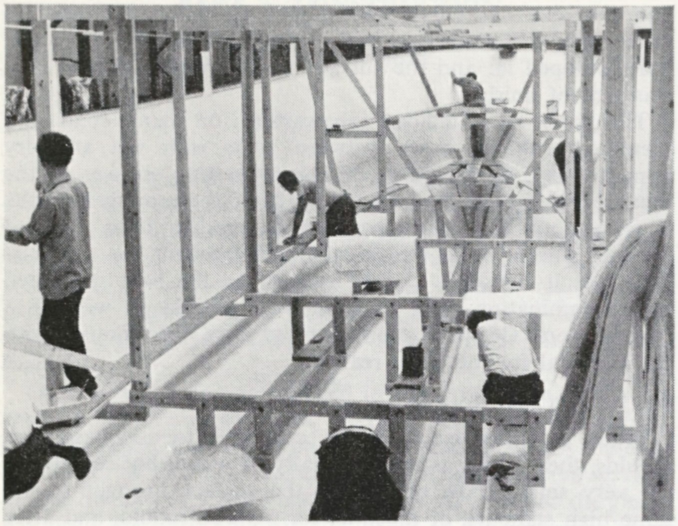


FIG. 2
Hand lay-up in progress from internal staging.

As soon as this operation is completed, the mould should be covered with an overhead canopy, extending beyond its boundaries to prevent any dust from settling upon the waxed surface. Polythene sheeting is often used for this purpose as it allows light to enter the working area.

3 . 3 . 2—PREPARATION OF THE RESINS

The next step is to prepare a sufficient quantity of gel coat resin to completely coat the mould. These resins are usually specially formulated with sufficient flexibility when cured to stand up to any flexing that the moulded hull may undergo in service without crazing or cracking.

When large areas of vertical surface have to be coated, as in the topsides of marine craft, it is usual to add a thixotropic agent to the resin to prevent drainage, but care should be taken to add only just sufficient of this to keep the resin suspended while setting.

Where it is desired to colour the resin, care must be taken to select pigments that will not upset the cure of the resin and in this respect carbon black must be avoided like a plague. Special polyester pigment pastes are now marketed comprising suitable pigments milled into a polyester resin and these are the safest types to use. Some of these, however, can affect curing times and tests should be carried out with the various colours in order to arrive at the right proportions of catalyst to be added to arrive at the correct setting time.

Now that the myth about plastics craft never needing painting has been exploded, a number of firms have abandoned the practice of colouring the resins, and are painting the surface of the hulls with polyurethane, coloured to suit the customer.

When the appropriate amount of resin has been measured out, the catalyst is well stirred into it, the thixotropic material added and stirred in, and the resin allowed to stand in the mixing bay to release any air bubbles trapped during the mixing process.

The accelerator is added to carefully weighed amounts of the catalysed resin just prior to it being issued out to the laminators, and is preferably handed out in shallow containers to reduce the build up of heat. The amount of accelerator added should be such as to give a setting time of about 45 minutes, for if too long a setting time is given, some of the styrene in the resin will evaporate and so give an undercured surface.

3 . 3 . 3—APPLICATION OF THE RESINS

The gel coat is applied to the mould surface either by spray, brush or lambswool roller in one operation. The roller method is the most foolproof, is quicker than brushing and produces an even spread. Spraying requires a high degree of skill and is considered too risky to use for gel coat application.

Surfacing tissue is sometimes used to reinforce the gel coat as it holds more resin than other types of glass fibre reinforcement. This material is known in the trade as "Candyfloss" and it is just about as useful as its nickname indicates, for the weather surface of yacht hulls.

As soon as the first gel coat has set, a second gel coat is applied and when this is set, the laminating proper can begin.

3 . 3 . 4—LAMINATING THE HULL

This consists of laying up glass fibre reinforcement, impregnated with resin, layer upon layer, until the required thickness is reached.

In hulls for marine work, this reinforcement is usually chopped strand mat, although some firms use a combination of woven glass cloth or woven rovings and chopped strand mat. Woven cloths and woven rovings can produce laminates of high tensile strength, but when placed adjacent to each other, interlaminar cohesion can be poor, so if woven materials are to be used, it is best to alternate them with chopped strand mat. One firm, with a view to backing up the gel coat and protecting the chopped strand mat against chafe, follows the gel coat with a scrim cloth of 6 oz. weight before laying-up the remainder of the hull in chopped strand mat, and here it is perhaps as well to note that in cases where a plastics hull is subjected to abrasion or chafe, that once the gel coat is rubbed through and the chopped strand mat exposed, chafe resistance drops rapidly, so there is a lot to be said in favour of backing up the gel coating with something better than "Candyfloss" or surface tissue.

The technique of laminating by the hand lay-up process is to apply a liberal coat of resin first, lay the glass reinforcement upon it, then to apply the remainder of the resin to the glass reinforcement. This soon dissolves the binders holding the glass reinforcement together, and if the resin is applied with lambswool rollers, the glass reinforcement is quickly rolled down into its position, and before the resin sets up, each layer of reinforcement is consolidated with a metal roller. Metal rollers can be obtained in various patterns, some resin manufacturers advocate the "paddle wheel" type, others the multiple washer or "parsley cutter" type, but those made up from solid stock with concentric grooves turned in them take a lot of beating, for their heavier weight quickly rolls out any air bubbles, consolidates the laminate, and even on chopped strand mat reinforcement, produces an acceptable finish to the inside of the hull.

It is important to control the ratio of resin to reinforcement to give a resin./glass ratio of $2\frac{1}{2}:1$ to $3:1$ and this is achieved in the hand lay-up process by issuing a given weight of resin to the individual laminators for each square yard of reinforcement.

As woven cloths and chopped strand mat are normally marketed in rolls 3 ft. wide, the normal practice is to hand

up 6 lb. of resin for every 2 yards of 2 oz. chopped strand mat, as this gives the right proportion of resin to glass and is a convenient amount of resin and glass for each laminator to handle at a time. This method may appear crude to the very scientifically minded, but it works well in practice and is simple. Laminating operatives are usually simple folk, if they weren't they would be engaged in a job less messy, less smelly and more remunerative.

When placing the reinforcing material, the edges are lapped $1\frac{1}{2}$ in. to 2 in. and the lap positions are suitably "shifted" on succeeding layers so as not to coincide.

It is not possible on a large hull to complete the laying up of the shell in one operation, but each layer should be completed and it is quite permissible to allow up to 18 hours if necessary between the lay-up of successive layers, but after this time, the layer that has set should be rubbed down with abrasive paper before the next layer is commenced.

When the shell is laid up to the required minimum thickness all over, extra layers are laid in the positions called for and the Society's recommendations in this respect are in the centreline, in the bottom up to the bilge, and at the sheerline. The layers of reinforcement so added require to be gradually reduced to avoid hard spots, although it can be noted that fatigue tests recently carried out in the aircraft industry on notched specimens indicate that a well made glass reinforced plastics stands up better than metals in this respect.

3 . 3 . 5—Laminating the other Structure in the Hull

Once the shell lay-up is completed, floors, ribs, bulkheads, tanks, engine seatings, etc., usually partly or wholly prefabricated in another part of the moulding shop, are bonded into the hull. This gives the laminate longer curing time in the mould and a stiffer structure to handle when being removed from the mould.

Ribs can be laid up over metal, wood, polythene tubing or paper rope formers. One patented system is an ingenious roll of this light alloy, "Top Hat" section, as in Fig. 10(c), cut at close intervals to enable it to be laid around a curve to form the male mould upon which ribs or stringers can be layed up. Lengths of this are cut off the roll and placed in position inside the hull and held by suitably placed wood chocks stuck to the hull with resin.

Floors of the plate or diaphragm type can be sawn from lengths of suitable reinforced plastics section previously laid up in a separate mould, and bonded to the hull with flanges of glass mat.

Bulkheads are usually of resin bonded plywood bonded to the hull with resin/glass flanges.

To ensure an efficient bond, the boundaries of the plywood bulkheads should be suitably scored to afford a good key and sometimes a mechanical bond is made by drilling holes around the boundaries of the bulkheads and inserting glass rovings, the ends of which are laid into the glass mat used for forming the flanges attaching the bulkheads to the hull and bonded into it with the resin.

When the bonding of these items to the hull is completed, it can be removed from the mould for inspection and then transferred to the finishing off section.

If the hull is a large one, it is usual to place it upon a wheeled cradle already prepared to receive it so that it can be easily transferred.

3 . 3 . 6—Hull Inspection on Release from Mould

The hull should be examined for flaws and surface imperfections and these items should be made good, as the materials for doing so are readily available in the moulding section.

The outer surface should be examined by sighting along the surface rather than directly on to it for local undulations or blisters can be quickly spotted and ringed in pencil for attention in this way.

These defects are usually the result of poor adhesion between the gel coat and the laminating materials through inadequate consolidation during the laying-up process. All loose resin and reinforcing material in these areas should be cut out, cleaned and dried and activated resin applied to the area and allowed to set. The resin should be liberally applied and held in position with cellophane secured with adhesive tape. Where the area is small the cellophane will give a smooth finish to the surface and no further treatment will be necessary, but if the area is large, then it will be necessary to rub the set resin down smooth with the surrounding area and finally polish it with a polishing compound.

Tapping at close intervals with a coin on edge will indicate the positions of air pockets or resin starved places in the laminate, caused usually by insufficient wetting out or consolidation, and these must be cut out and made good with resin and reinforcement.

3 . 4 FITTING-OUT AND COMPLETION

3 . 4 . 1—Tank Testing

At this stage, unless the tank structure has to be drilled to receive ballast keel bolts, or engine foundations, integral fuel and fresh water tanks should be pressure tested to ensure they are tight and in cases where the tests have of necessity to be carried out in the moulding area, the Society allows air pressure to be used in place of hydraulic pressure.

As normal air pressure gauges can be very unreliable at the relatively low pressure required, 3 to 5 lb. per square inch, it is better to use a test rig on the lines shown in Fig. 3. The pressure should be maintained for at least six hours to safeguard against loss of pressure through osmosity of the reinforced plastic material.

3 . 4 . 2—Fitting the Ballast Keel

In sailing yachts where ballast keels are bolted to the hull, it is usual to mould into the hull a strong backbone member reinforced with steel rods as in reinforced concrete structures and metal tubes of the same material as the keel bolts are moulded into this backbone member through which the bolts pass. The bolts should be bedded in resin where they pass through the tubes, and after being hove up tightly, it is good practice to cover the nuts with a resin/glass cap to ensure against leakage.

A fairly recent innovation is to dispense with keel bolts and to place the lead inside the moulded hull, either in the form of precast pieces of lead or particles, and to bond this in with resin and glass mat.

The Author must confess to some misgiving over this method of ballasting not only from the point of view of the stress placed upon the hull when the yacht is heeled, but from the point of its being able to stand up to the standard method of removal from the cradle when being laid up for the winter, e.g. blocking up and lifting clear of the cradle with driven wedges. However, time alone will give us the answer to this

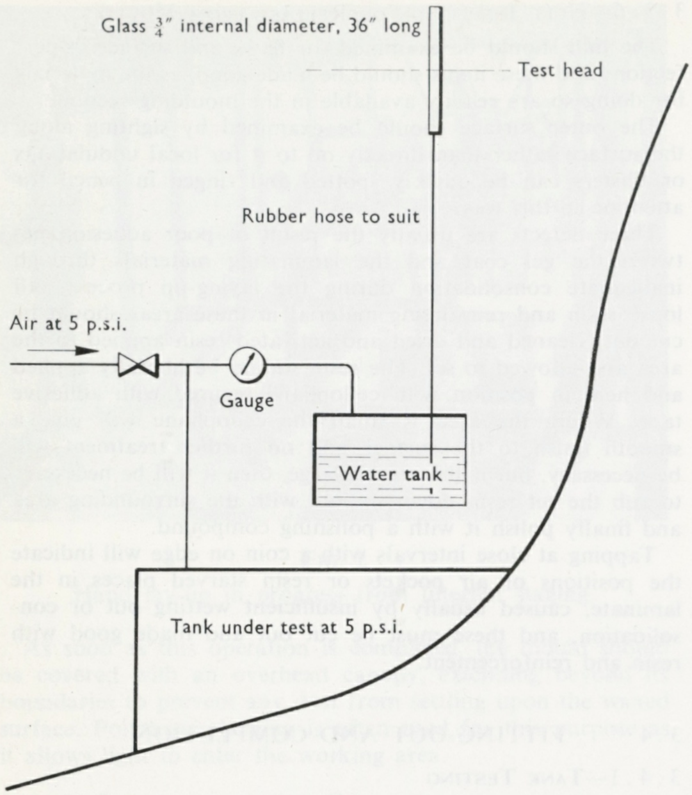


FIG. 3

Layout of air pressure test rig.

and in the meantime it is of interest to say that one yacht constructed in this manner grounded on a rock on her maiden voyage, listed over on to another piece of rock that happily supported her with not too great a list whilst the tide dropped 30 ft. and rose again in due course to enable her to be floated off. She remained perfectly watertight and, apart from some indents and scores to her forefoot and underside of keel, which were repaired in two days on her return to this country, she escaped undamaged.

3.4.3—ATTACHMENT OF THE CHAINPLATES

Chainplates and anchorages for the lower ends of other standing rigging require special consideration in the design of their attachment to the hull and as no reliable data has yet been devised for assessing the stress that these anchorages have to withstand, perhaps a good rule would be to place the figure at twice the breaking stress of the respective items of rigging wire anchored to them. Further, as we also have no reliable data upon which to calculate the strength of metal anchorages to resin/glass laminates, it seems the logical procedure to make up trial samples of the proposed anchorages on pieces of resin/glass laminate and test them to destruction, or at least to double the breaking strength of the wire rigging attached to them and believed to be the normal practice of reputable designers and/or laminators.

These anchorages can vary from the bolted-on type with the hull thickened up in way of the bolts, to tapped receiving plates bonded into the moulded hull, or even a combination of both these means, but the golden rule would appear to be to design the moulding in such manner that the stress exerted by the rigging to the anchorages is spread over a large area of the hull and this is easily done by adding extra layers of chopped strand mat or woven rovings to thicken up the laminate.

3.4.4—INTERNAL JOINERWORK

After the deck unit is fitted, the inside furniture can be fitted in the hull, and as resin/glass mouldings present a new problem to the average marine joiner, from the fastening point of view, the attachment of inside furniture should be properly thought out and drawn up before the joiners are allowed loose aboard, for at least one case is known of a rib fracture occurring due to a joiner attaching side linings with self-tapping screws direct to the inside face of the ribs, although in this case, the joiner made a bad shot with the first hole and drilled a second hole alongside. Subsequent test made by the moulders showed this reduced the strength of the rib by 50 per cent.

Wherever possible, internal joinery should be fastened to the wood bulkheads, but resin/glass boundaries can be moulded into the hull to carry the ends of wood floor bearers, lockers, fronts, etc., provided their respective positions are determined in good time.

Wood grounds can also be bonded to the hull to which side linings can be screw fastened without weakening the hull structure.

3.4.5—COMPLETION

After the internal joinery and installation of the machinery is completed, it is simply a matter of rigging and equipping the craft and putting into commission and as this follows the same general form as that for wood craft, it would be superfluous to deal with these matters in this paper, except to state that any fittings that are of necessity bolt-fastened to the plastic hull must be given special treatment to ensure watertightness, for when bolts are driven into wood, the holes can be driven to fit, whereas in reinforced plastics they must be a push fit, so it is desirable to coat the bolts with resin and to fit a resin impregnated glass fibre grommet under the heads of the bolts before they are hove up.

PART 4—CONSTRUCTION DESIGN

By A. McInnes

4.1 INTRODUCTION

The methods of constructing plastics boats are conspicuously different from those used in building boats from conventional materials. In plastics manufacture, basic raw materials in the form of liquid resins are brought together with glass reinforcements to form a finished or nearly completed structure in a few simple operations. Construction methods and techniques have undergone continual change since they were introduced into this country. Full exploitation of the use of this material is, as yet, far from being achieved.

The two basic boat construction techniques described have been developed for building the majority of plastics boats to-day. In the early days almost all craft were of single-skin construction and now, due to the development of suitable cores, sandwich construction will become more common. The single-skin hull formed in a female mould is the most advantageous where a number of boats are to be built and the sandwich construction is more adaptable to custom building methods. Because of these advantages, both types of

construction are likely to continue in use in the near future. It is considered that single-skin construction will maintain its present position in hulls up to about 70 ft. in length that are to be moulded in any numbers and that new developments in sandwich construction will be used for the custom built craft and production craft above this length. This paper, therefore, deals primarily with single-skin construction, although many of the remarks are also relevant to sandwich construction.

4.2 TYPES OF CONSTRUCTION

4.2.1—Types of Construction

The hull of a plastics boat may be a moulded single-skin laminate, sandwich construction of thin laminates with a low density core, or some combination of these types. The selection of the type of construction is dependent on hull size, intended service, yard capabilities, production facilities and economic considerations. Some doubt may exist as to the most suitable type of construction for any specific boat hull. Usually more than one type of construction is suitable, but only one specific type will result in the most economical design.

Generally, the builder, in developing a new small boat for quantity production, will produce a prototype to test the soundness of the hull, performance of the hull shape and suitability for series production. Modifications to improve the original prototype and reduce production costs are generally made prior to and in the early stages of a series run. This is usually impractical for the larger boat unless quantity production is contemplated.

4.2.2—Single-Skin Construction

The majority of plastics hulls are moulded using the single-skin construction which comprises a laminate of several layers of reinforcement, moulded to the desired form either "unstiffened" or "stiffened".

The term "unstiffened" as used here means that no stiffening sections are added and that the hull has an interior surface unbroken by protruding frames. Often interior assemblies, such as thwarts and buoyancy tanks are used to provide some support to the hull laminate in addition to performing their primary function. This type of construction derives considerable strength from the curved shape common to most small boats and is normally used in open, slow-speed boats, small sailing yachts and in some lifeboat designs.

As the size of the boat and the severity of service increases or when large flat surfaces are required, the unstiffened single skin becomes too flexible and rigidity has to be obtained by the addition of stiffeners. This could also be done by large increases in the laminate thickness, but would be very uneconomical. The stiffeners in this case may be a stiffening section without any further purpose, as described in Section 4.4.3, or they may be a built-in bunk, locker, or other component already serving an additional function.

The framing is usually orientated in two basic directions, longitudinal or transverse, and the choice of orientation depends largely on a compromise between the interior arrangement and economical construction of the hull. Longitudinal framing is frequently preferred in the boat below 40 ft. as much of the internal structure can be used to support or replace the longitudinal frames. In the bigger boat, however, a complete framing system independent of any internal fittings is used and there appears no clear preference for either system. Some of the advantages of having a completely framed hull are that it allows freedom of internal layout in a standard hull and that the internal joinerwork does not need to be structurally matted-in, an important consideration where hulls are transferred out of plastic shops for completion.

4.2.3—Sandwich Construction

The sandwich is the most complex type of construction and the most difficult to fabricate. It consists of two plastic laminates separated by a core of lightweight material, the purpose being to increase the rigidity of the flat panel by increasing its thickness without the use of the solid laminate.

In sandwich construction it is usually assumed that the thin face laminates resist all the bending stresses and deflections, while the core resists the shear stresses and deflections, withstands local crushing loads and prevents buckling of the face laminate in compression. Since the strength and rigidity depend on both faces working as a unit at the required separation, the core material must bond firmly to each skin and be sufficiently strong to withstand the above loads.

The effectiveness of the core on the overall strength and rigidity of the sandwich varies widely with different physical properties of the core. Some cores, notably wood, have sufficiently high flexural moduli to cause an appreciable effect on the flexural rigidity of the sandwich. Others have such a low shear modulus that the effect of shear deflection must be taken into account.

The most common core materials being used are balsa wood, expanded plastics, foamed resins and honeycombs. Balsa wood and marine plywood have proved satisfactory but experience has shown that solid hardwood cores, encased in the laminate, have a tendency to swell and thereby crack the covering laminate. Rigid closed-cell expanded plastics, particularly polyurethane, cellular cellulose acetate (cca), p.v.c. and phenolic are used, as is polystyrene, but this latter material must be coated to prevent contact with resins containing styrene. Honeycombs made of heavy duck and resin impregnated paper are satisfactory, but must be handled with extreme care to ensure complete resin impregnation to obtain a good bond between the core and the faces. Another useful material is "microballoons" which is a mixture of lightweight gas-filled phenolic spheres embedded in either polyester or epoxy resin. This material forms a denser core which can be easily trowelled into awkward corners.

Because of the great increase in stiffness and load carrying capacity, sandwich is frequently used for relatively large flat panels, particularly where the presence of stiffening sections would be undesirable for internal arrangements. When the shape is complex, the pre-formed cores are generally difficult and expensive to shape. In general, therefore, it is considered most suitable for use in decks, cabin tops, and interior bulkheads, but may be troublesome for the hull laminate. This is partly due to the fact that in the event of damage, the repair is more difficult than in single-skin construction, and to the difficulty of ensuring and maintaining an adequate bond to the adjacent parts for necessary continuity.

4.2.4—Composite Construction

Composite construction in the truest sense, namely, a single-skin plastics hull with wood or metal framing, is now comparatively rare. This method was more common in the early days when conventional wood framing was often screwed or

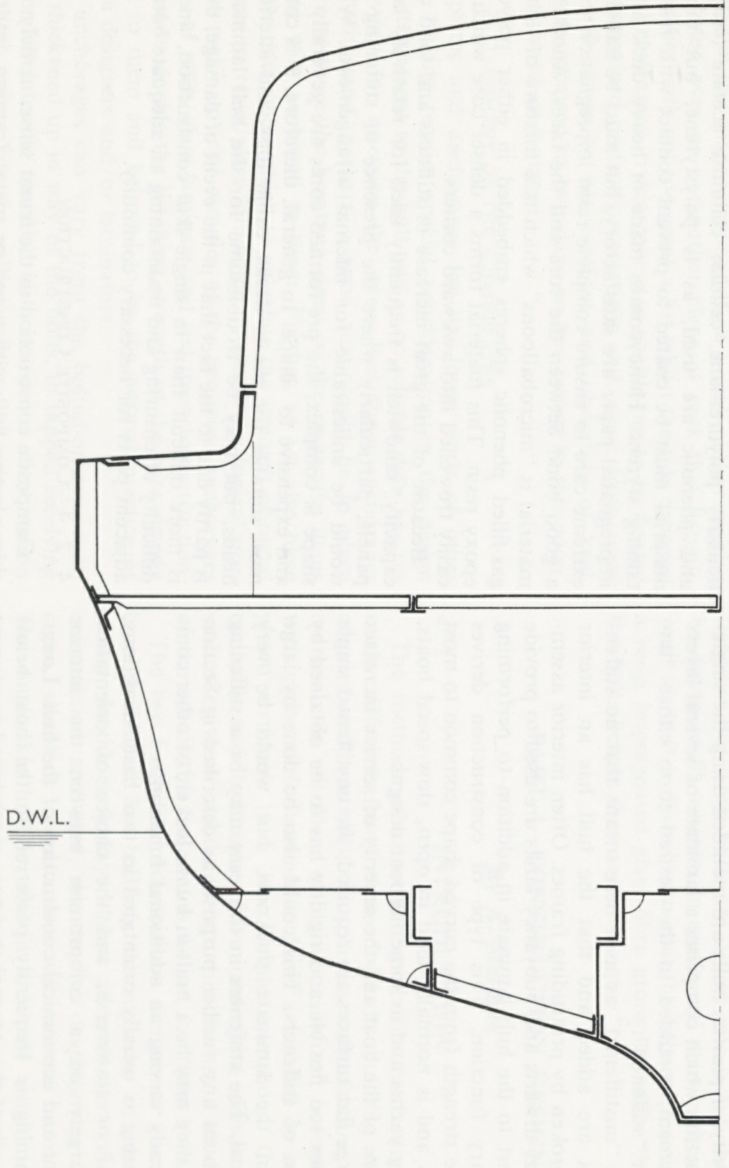


FIG. 4

Midship section of motor yacht single-skin construction.

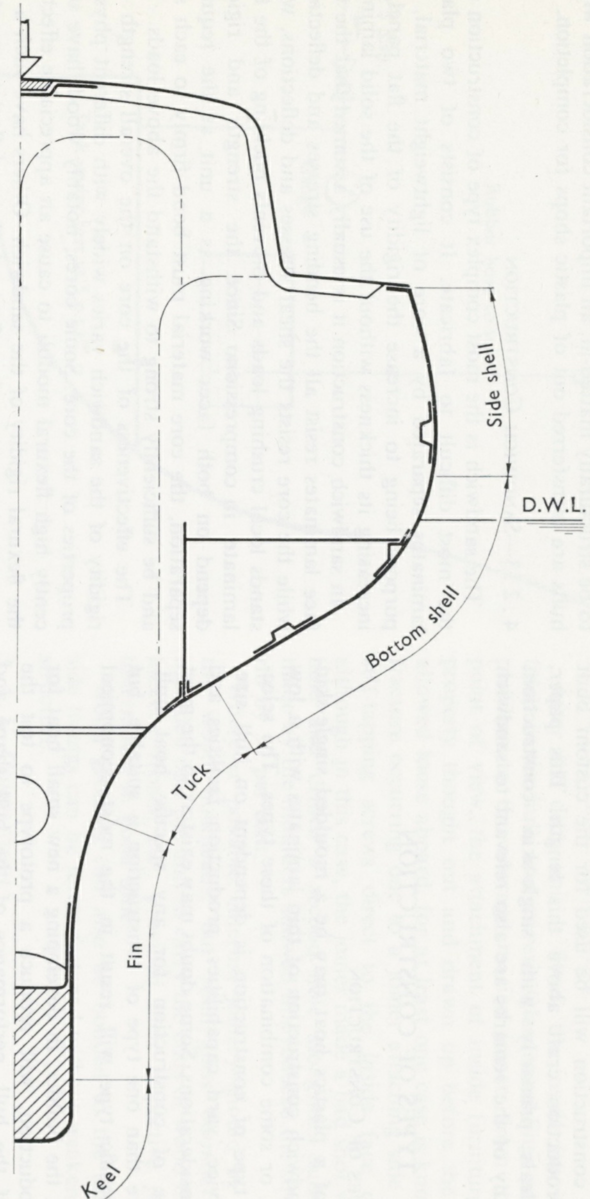


FIG. 5

Midship section of sailing yacht single-skin construction.

fastened to the hull laminate or aluminium alloy frames and bottom girders were used to stiffen the hull. Nowadays "composite" is used more loosely and generally refers to the fitting of a conventional wood deck to the all-plastics hull or the encasing of wood or metals into parts of the plastics structure.

This encasing of wood and metals is one of the major sources of controversy among designers and builders in view of the various difficulties experienced with these materials. Encased woods may swell and crack the plastics covering, or rot and become incapable of supporting loads. Encased metals may rust or corrode and cause failure of the plastics to metal bond. Encased foams may crumble under impact or fail in shear under high local loads. Failures are extremely difficult to detect because of the plastics covering and subsequent repairs are very costly.

Another source of trouble, due to the basic properties of the materials, is the large difference in moduli of elasticity and the unbalance in a structure of the metals and plastics. This modulus for aluminium is about ten times greater and of steel 30 times greater than that of a mat laminate. If a piece of steel is bonded to a laminate, both will deform together under the load and the stress in the steel will be about 30 times that in the plastics, and as the failure stress of steel is nowhere near 30 times the failure stress of plastics, the part may fail due to rupture of the steel before the plastics has reached its ultimate strength.

Due to the possible difficulties arising from corrosion of the metal or failure of the metal-plastics bond, plus the problem of the hard spots created at the ends of the steel material and the possibility of high stresses due to temperature change, the use of encased metals must be watched carefully and be subject to tests before being incorporated in structural members under production. The use of steel and other metals as local members for distributing a concentrated load to a wider area of plastics is, under proper conditions, considered good practice as the materials are not being stressed together as when encased.

4.3 DESIGN CONSIDERATIONS

4.3.1—Structural Considerations

Plastics structures can be designed on either the "flexible" or "rigid" theories.

The plastics hull is a one-piece monocoque type structure with greater inherent strength, tightness and flexibility than its wood counterpart. The high strength of the laminates and their ability to give under load permit the use of flexible shells of minimum weight. This theory requires extreme care in application, for if large deflections are permitted in the laminates, the need to avoid hard spots becomes substantially more important, since their adverse effect increases with increased deflection. The flexible design approach is only used when it is possible to avoid hard spots completely, such as in small slow speed round bilged craft.

The Society's Provisional Rules for the Construction of Reinforced Plastic Yachts are based on the rigid theory and result in greater thickness of laminates and stiffening sections with relatively small deflections and low stresses under load. The effect of the low stiffness modulus can be readily seen in flat unsupported panels which are, of course, highly resilient and will return to their original shape even from extreme deflections. Panel flexing is not unacceptable provided the resultant strains are transferred and relieved. But flexibility in the structure should not be accepted beyond its functional point, such that decks are firm and secure underfoot, bottom panels should be stiff enough to give a planing surface free from distortion at the designed speeds and loadings, and the topsides should be rigid enough to withstand rough docking manœuvres or similar operational conditions.

An important advantage is that the material can be easily moulded in shapes which can increase the panel stiffness to a very large degree. A hull form with plenty of curvature and a generous rounded bow with a graceful flair can easily be adopted. Inherently stiff structural shapes such as spray chines, sheerstrake knuckles and skegs can be readily achieved as can variations in hull form, such as tunnels, etc. Decks and deckhouses are designed with well cambered decks and well radiused corners, and often small recesses and local compound curvature can be adopted.

Stress concentrations from bulkheads, frames and other attachments do not result in excessive stress levels. This, however, does not allow those connections to be made carelessly and hard spots should be avoided where possible and when necessary their effect should be minimised by attention to detail design. Attention should be similarly paid to change of laminate thickness to ensure that there is a gradual transition from one thickness to the other, particularly in areas susceptible to vibration.

4.3.2—Selection of Laminates

The choice of type and arrangement of reinforcement for a single-skin laminate is based on a number of factors. These factors include strength, rigidity, impact resistance, resistance to passage of water, cost of material and labour, ease of handling and appearance. Each of the basic types of reinforcement has its own qualities which must be utilised to provide the most effective laminate.

There are obviously numerous possible combinations of reinforcement that may be used and it is proposed to mention only three of the more common types and discuss their relative merits.

The first laminate comprises the basic chopped strand mat with perhaps a layer of scrim cloth or tissue on the outside to reinforce the gel coat. This is the standard laminate used in the United Kingdom and is also used by some Continental boatbuilders, particularly those constructing lifeboats. Its main advantages are that it gives the maximum economical build-up and is easily handled and moulded into complex shapes. The mat laminate also has good resistance to water absorption and good interlaminar bond, both of which reduce the extent of damage and simplify repair. As can be seen from Fig. 6, the mechanical properties are not very high but are adequate for the purpose in these small craft.

	Chopped strand mat—2 oz.	*Woven roving 25–27 oz.*
Glass content %	30	50
Specific gravity	1·42	1·64
Thickness per ply ins. ...	0·055	0·037
Tensile strength—p.s.i. ...	14,000	33,000
Tensile modulus—p.s.i. ...	$1{\cdot}0 \times 10^6$	$2{\cdot}1 \times 10^6$
Flexural strength—p.s.i. ...	21,500	27,100
Flexural modulus—p.s.i. ...	$0{\cdot}95 \times 10^6$	$1{\cdot}81 \times 10^6$

Fig. 6

Typical mechanical properties.

Two other laminates of interest are American practice and are also used by several Continental boatbuilders, mainly those building for the American market. The first consists of a layer of 10 oz. (350 gr.) boat cloth on the outside, one layer of 2 oz. (600 gr.) mat and a varying number of layers of 27 oz. (900 gr.) woven roving. The cloth layer on the outside is intended primarily for appearance and to reinforce the gel coat; its relatively smooth surface tends to reduce the gel coat thickness and also imparts certain good tensile qualities on the exterior of the laminate. Some builders, however, may use a 1 oz. (300 gr.) mat in place of the cloth. The inner layer of mat serves as a barrier against passage of water into the laminate and also prevents the weave of the woven rovings from showing through the cloth and the gel coat. The second laminate is a better proposition and differs in that the bulk and strength of the laminate are provided by layers of mat in place of the woven roving. One or two layers of woven roving are used on the inboard face to provide the tensile properties necessary for good impact resistance.

It is hard to justify the use of woven rovings in these small craft under consideration unless minimum weight and high impact resistance are major requirements. Although such laminates have a much higher impact resistance than the mat laminates, when they are damaged, the damage is often more extensive due to failure of interlaminar bond and the effect of water absorption makes the repair more difficult. The interlayer bond between layers of the heavier roving fabrics, about 27 oz. (900 gr.), is often not very good and can be improved by incorporating a layer of mat between the woven roving layers. Some builders in the United Kingdom use a lighter weight of woven roving fabrics, between 12 oz. and 18 oz. (400 gr. and 600 gr), which are much easier to wet out and handle, and in which good impregnation can be guaranteed. These lighter fabrics should be used in the face laminates of sandwich construction where the prime object is to produce a laminate which is light and rigid.

Apart from imparting good tensile properties to the inner face, woven rovings and cloth are often used to improve the inside appearance of the hull and deck structures.

4.3.3—Production Considerations

The moulding method is invariably the contact process, either by hand or spray lay-up, and this, with the type of construction, will determine whether a female, male or batten mould is to be used. The size of production will determine the mould material. The majority of plastics boats are built in a female mould and often comprise only two or three components, the hull, and deck with or without the superstructure. There are, of course, many subsidiary moulds to produce floors, tanks, bulkheads, hatches, etc.

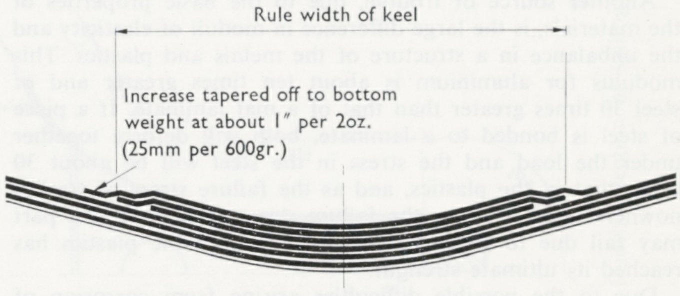
The hull mould is spit into convenient sections which are normally along the centreline and the transom boundary. The deep fins in sailing yachts may be further split into two or three sections to permit access for laminating. Often this mould is arranged to split and give variations in hull design such as a longer hull, deeper fin, single or twin screw tunnel arrangements, etc. The builder may prefer to break the superstructure up into several parts, perhaps for shop handling reasons or to offer alternate model arrangements.

The laminating and assembly procedure is fully worked out at the design stage as time is vital once the lay-up has commenced and should not be wasted sorting out problems that could have been foreseen. This is known as the "phasing", and means ensuring that each layer, item and component is laminated in its proper sequence such that either primary or secondary bonds are always achieved and also that no bonded joints, apart from those designed as such, have to be resorted to. The structural details are examined to ensure they are feasible and easily moulded, and if complicated are redesigned, either in plastics or conventional materials. Conventional materials may also have to be used to avoid over-stretching the boatyard's plastics capacity and labour force.

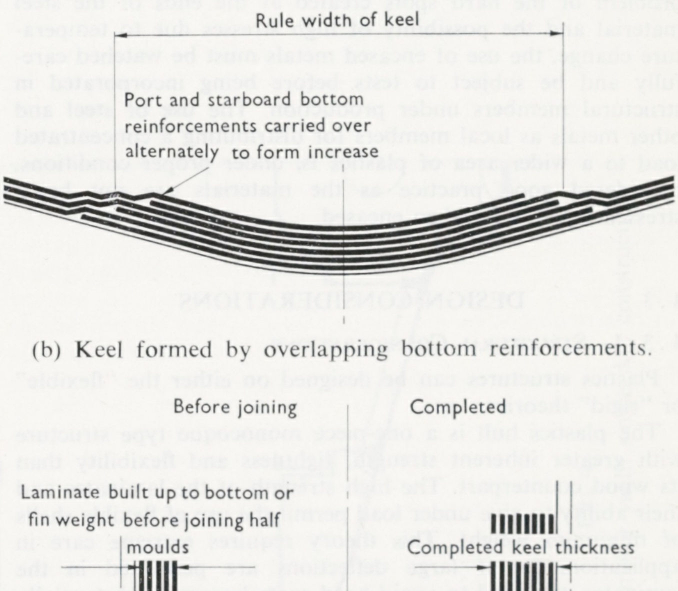
4.4 HULL CONSTRUCTION

4.4.1—Hull Laminate

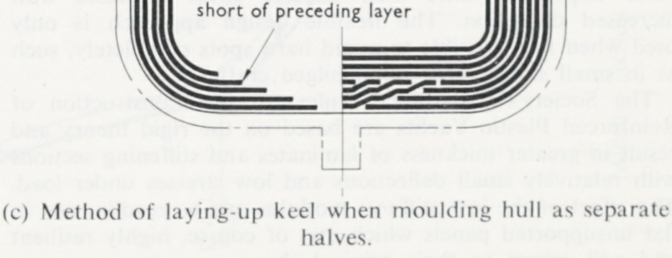
The weights of the hull laminate on the sides and bottom of the boat are given in the Rules and are related to a basic stiffener modulus at a given spacing. The weight of the



(a) Keel formed by additional layers of material.



(b) Keel formed by overlapping bottom reinforcements.



(c) Method of laying-up keel when moulding hull as separate halves.

Fig. 7

Various configurations of keel lay-up.

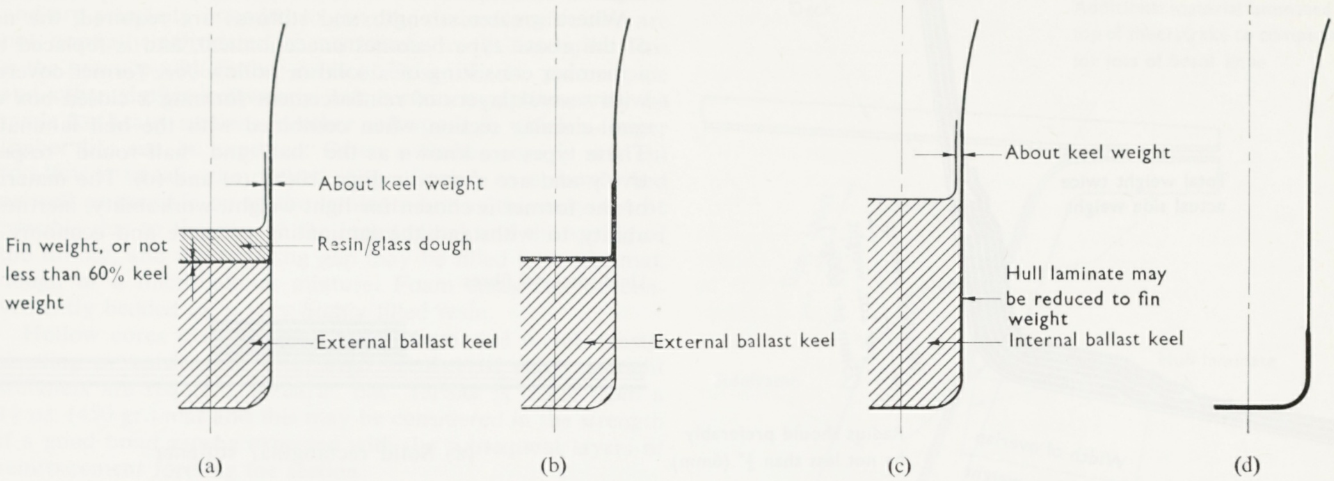


FIG. 8
Variations of keel lay-up in sailing yachts.

laminate in these areas can be modified for differences in stiffener spacing and for differences in stiffener modulus.

A keel is to be formed by laminating additional material along the centreline of the hull and is to extend forward to the stem head and aft to the transom boundary, so forming a structural backbone. This backbone can be maintained at the midship width throughout the length, and be reduced in weight towards the ends, or can be constant in weight and reduced in width, or can be a combination of both. This flexibility is to suit whichever method is used to obtain the backbone increase as this can be done in various ways. In Fig. 7(a) the increase is formed by laminating additional layers of reinforcement along the backbone whilst in Fig. 7(b), where the hull is laid up transversely, the increase is obtained by graduating the overlap of the various layers of the port and starboard bottom reinforcements.

Where a deep skeg is incorporated in the hull form in the larger motor yacht and in the fin and tuck areas in sailing yachts (*see* Fig. 5), the weight of the laminate is usually intermediate with that of the keel and bottom. This laminate is normally increased if the floor spacing exceeds twice the rule basis spacing. In some sailing yacht designs, to suit production, a heavy bilge strake is formed by overlapping, or butt strapping, the bottom and tuck reinforcements.

In sailing yachts, particularly those with fine deep fins, laminating in this area is difficult due to the restricted and uncomfortable access. It is quite common, therefore, for the hull to be moulded in halves, each half being completed to just exceeding bottom weight depending on the phasing, then the mould halves are bolted together and the laminating completed. Fig. 7(c) shows the scarphed joint where each layer is stepped back a minimum of 1 in. (25 mm.) from the preceding layer. This method has the additional advantage that the half mould is in such a reclining position that the moulder can generally operate from the shop floor and can therefore work much faster and also achieve a slightly lower resin to glass ratio.

Figs. 8(a) to 8(d) show various keel configurations in sailing yachts. Figs. 8(a) and 8(b) indicate the method which was standard practice several years ago in which an external ballast keel was bolted to the hull. Some cautious designers used a heavy keel section, formed by a resin glass dough between the keel laminates, to take the loading from the keel bolts. The latest concept is to fit an internal ballast keel and carry several layers of reinforcement over the top of the ballast to tie the fin sides together and so compensate for the loss of floors, as in Fig. 8(c), while clear of the ballast, the laminate is as shown in Fig. 8(d).

The topsides are increased to form a sheerstrake in motor yachts and further increases may be required in high speed craft and in special service boats which are frequently coming alongside quays or other vessels. In sailing yachts the topsides in way of the mast are increased to take the shroud loads, but this is dealt with in more detail in Section 4 . 7 . 2. The foregoing increases are also considered in conjunction with the deck edge connection (*see* Section 4 . 5 . 2) as the necessary material may be embodied in this connection.

The boundary of the transom is to be increased to effectively support and stiffen the sides, bottom and transom laminates. Fig. 9 shows a typical boundary lay-up formed by overlapping the side and bottom with the transom reinforcements, but as with the backbone, the increase can also be achieved by the addition of strips of material laid around the boundary. The weight and width of the increase are greater in full power craft than in sailing yachts as protection against quay contact on the sides and the additional loading and vibration from the stern gear on the bottom. In hard-chine boats, the hull is similarly reinforced along the chine line and this is usually done by overlapping the side and bottom reinforcements.

4 . 4 . 2—HULL STIFFENING

In very small craft, as mentioned in Section 4 . 2 . 2, the hull stiffening is obtained from the very rounded form and from internal assemblies such as bulkheads, thwarts and buoyancy tanks, etc. These items are usually matted-in to the hull with connections similar to Fig. 20(a). In sailing yachts up to about 40 ft., and in small motor boats, the stiffening provided by the internal assemblies, which now include bunk bottoms, shelves and other cabin furniture, may be supplemented by an occasional transverse or longitudinal stiffener. However, in the larger yachts and in quite a number of sailing designs

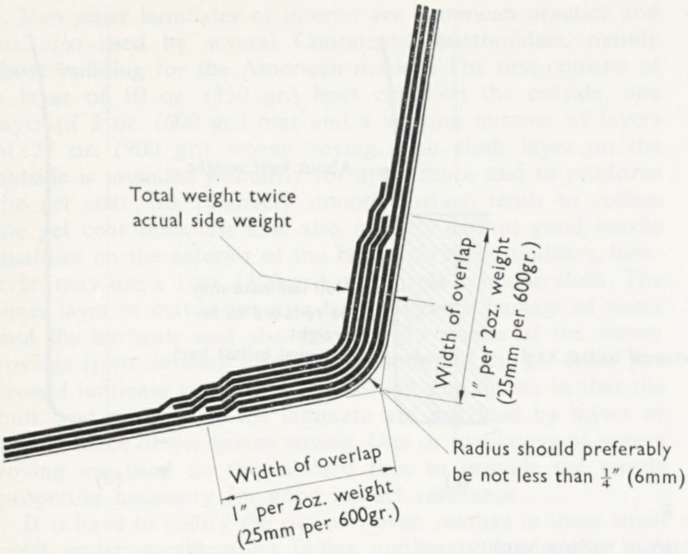


FIG. 9

Typical lay-up along chine and transom boundaries.

below 40 ft., a complete system of transverse or longitudinal stiffeners, or a combination of both, is arranged in conjunction with bulkheads and bottom girders but independent of the internal joinerwork assemblies. These stiffeners are usually some of the configurations described in the Section 4 . 4 . 3.

Floors are usually fitted in the fins of sailing yachts and in the skegs of the larger motor yachts. They are usually spaced about twice the basic transverse frame spacing and may be every frame space in fuel tanks. Floors, at similar spacings, are normally fitted on the bottom forward in the motor yacht, but usually only one or two are fitted in the slamming area in the sailing yacht. It is usual to prefabricate the flanged floors on a bench, cut to shape, fitted and matted-in using the standard connection shown in Fig. 20(a).

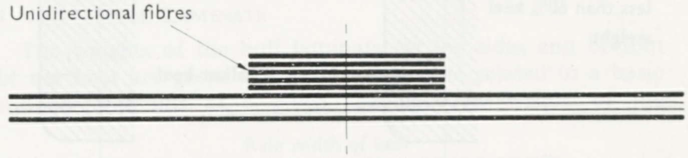
Floors may also be fitted on the comparatively flat bottom of transversely framed motor yachts, but it is more common to carry the framing section continuous over the bottom from side to side, and support the frames with bottom girders. These girders should align with the engine girders and extend from the transom to as far forward as practicable. In conjunction with the keel and skeg, they maintain the hull stiffness over the entire length of the shafting installation and so help to avoid any alignment troubles in service. The bottom girders may be of plastics, but are more commonly plywood, similar to Fig. 12(c), and should be effectively fastened to the main bulkheads.

4 . 4 . 3—Framing and Stiffening Sections

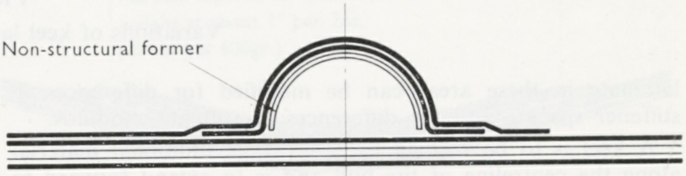
Typical configurations of stiffening sections which are used as frames, beams and panel stiffeners are shown in Fig. 10. The stiffener former is placed in its position on the laminate without undue delay and the section is built up layer by layer as a continuous process.

Fig. 10(a) is a solid rectangular cross-section built up from layers of unidirectional rovings, and with alternate mat layers if necessary to improve the inter-layer bond. This section can be easily laid up on the bench and laminated wet to the hull, or can be effectively bonded. This type is used in the small boat and in the larger boat in areas such as coachroofs, under engines, etc., where the section depth may be very restricted.

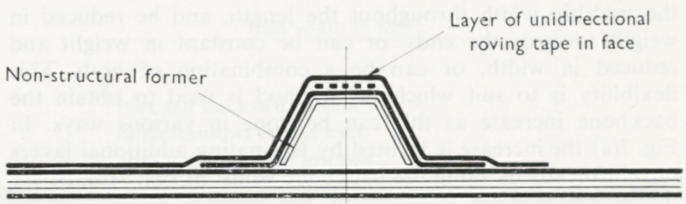
Where greater strength and stiffness are required, the use of the above type becomes uneconomical, and is replaced by a member consisting of a solid or hollow core former covered with several layers of reinforcement forming a closed box or semi-circular section when combined with the hull laminate. These types are known as the "hat" and "half-round" respectively and are shown in Figs. 10(b), (c) and (d). The material of the former is chosen for light weight, workability, inertness, ability to withstand the laminating pressure and economy.



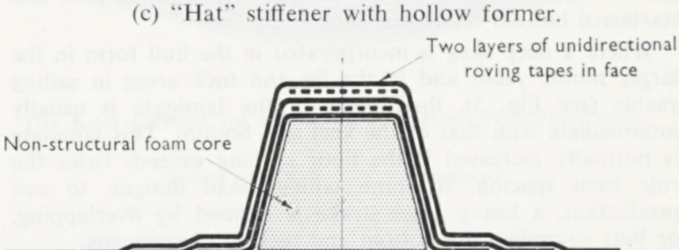
(a) Solid rectangular stiffener.



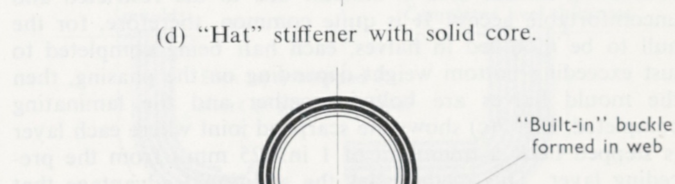
(b) "Half-round" stiffener with hollow former.



(c) "Hat" stiffener with hollow former.



(d) "Hat" stiffener with solid core.



(e) "Round" stiffener with hollow tube former.

FIG. 10

Typical framing and stiffening sections.

The solid cores are normally of foamed plastics or timber and are primarily intended to give the desired profile and are not normally considered to contribute to the strength. This is the practice with timber cores of balsa, obechi, parana pine and other similar softwoods, but certain exceptions may be made for heavy sections in other softwoods in which a longer timber life can be expected. Often a timber core is saw cut to allow it to take the desired shape. Such a core should bed as well as possible on to the laminate but as this may not be possible due to the run of the longitudinal and the spring of the timber, and the resulting gap may be filled with wet mat, dough or a microballoon mixture. Foam cores can be conveniently bedded on a very highly filled resin.

Hollow cores may be sheet metal, expanded metal, plastic sheeting or cardboard. If a large number of fairly straight stiffeners are required, often a "hat" former is made from a $1\frac{1}{2}$ oz. (450 gr.) mat and this may be considered in the strength if a good bond can be expected with the subsequent layers of reinforcement forming the section.

The strength and stiffness of the sections can be varied by adjusting the section depth and retaining a constant lay-up or by increasing the lay-up where the stiffener depth cannot be increased. The lay-up can be increased by additional layers of the same material or by incorporating high strength materials such as shown in Figs. 10(c) and (d) which include unidirectional roving tape between the layers of basic reinforcement. The basic reinforcement is chopped strand mat in United Kingdom and woven rovings in U.S.A. and Canada, whilst both reinforcements are commonly used in the rest of Europe.

The matting-in of the sections should be as neat as practicable with adequate overlaps of the basic material and the high tensile face material as continuous as possible. Sections, such as Fig. 10(e) using tubular formers such as paper rope are not recommended as the lay-up tends to have a "built-in" buckle, and requires careful attention from moulder in ensuring satisfactory shape. Often the last layer of the hull reinforcement is carried over the section, as shown on the left-hand side of the sections in Fig. 10, to improve the internal appearance by covering any untidy connections with their almost unavoidable resin runs.

Rather than finish in a small bracket as in steel construction, plastics stiffeners can often be tapered off at their ends provided they are run on to an area of increased thickness, as in Fig. 11. The lay-up material, particularly unidirectional tapes, should be neatly moulded into the heavy laminate to effectively transfer the load. This is of major importance as it allows work to be completed in its own mould and cuts down work which would otherwise have to be done on assembly to the hull when time is at a premium. Such small brackets, as beam knees and chine brackets, are laminated piecemeal and often misshapen and consequently their strength continuity and workmanship may be suspect.

4 . 4 . 4—BULKHEADS

Bulkheads play a greater part in a plastics hull than they do in wood or metal boats. Apart from their prime function of separating the various spaces, they are essential in providing transverse rigidity necessary for maintaining the form in the comparatively flexible plastics hull. There should be sufficient bulkheads or partial bulkheads to maintain the transverse strength of the yacht.

In yachts over 50 ft. waterline length, the Rules require a collision bulkhead and watertight bulkheads enclosing the

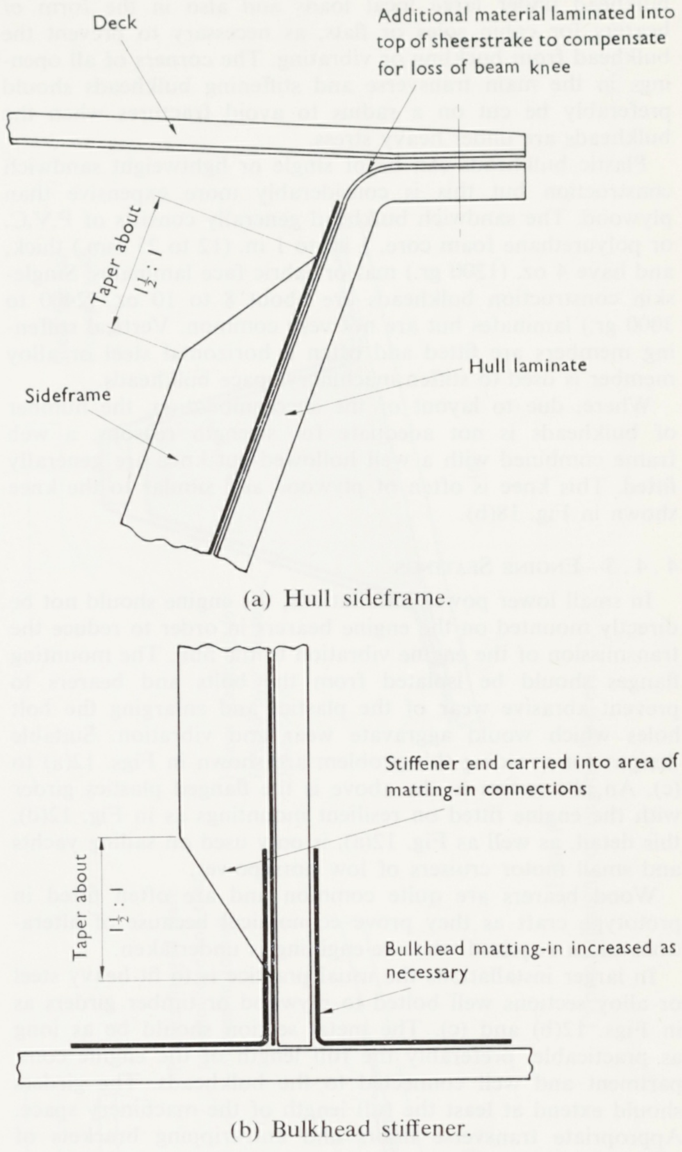


FIG. 11

Method of ending stiffening sections.

machinery space. In smaller motor yachts, the machinery space bulkheads only require to be gastight. The other bulkheads are not usually intended to withstand a water head but may be required to support the weight of equipment mounted on them, the loads imposed by the deck and side framing, and any local loads, such as from the mast.

Standard practice is to use marine plywood for bulkheads as it is cheap and convenient. They should preferably be fitted while the hull is still in the mould as this can be done more accurately and gives a better bond. Moreover, the bulkheads serve to strengthen and hold the hull rigid if it has to be transferred from the mould to another shop or another yard for fitting out. The matting-in connections shown in Fig. 20 are used, the thickness being increased for bulkheads under load, such as watertight and mast bulkheads. The plywood bulkheads are normally unstiffened in small craft but in the larger boats, stiffening, or pillaring, should be fitted on the

bulkhead under large local loads and also in the form of bearers for cabin soles or flats, as necessary to prevent the bulkhead from buckling or vibrating. The corners of all openings in the main transverse and stiffening bulkheads should preferably be cut on a radius to avoid fractures when the bulkheads are under heavy stress.

Plastic bulkheads can be of single or lightweight sandwich construction but this is considerably more expensive than plywood. The sandwich bulkhead generally consists of P.V.C. or polyurethane foam core, $\frac{1}{2}$ in. to 1 in. (12 to 25 mm.) thick, and have 4 oz. (1200 gr.) mat or fabric face laminates. Single-skin construction bulkheads are about 8 to 10 oz. (2400 to 3000 gr.) laminates but are not very common. Vertical stiffening members are fitted and often a horizontal steel or alloy member is used to stiffen machinery space bulkheads.

Where, due to layout of the accommodation, the number of bulkheads is not adequate for strength reasons, a web frame combined with a well hollowed-out knee are generally fitted. This knee is often of plywood and similar to the knee shown in Fig. 18(b).

4 . 4 . 5—Engine Seatings

In small lower power installations, the engine should not be directly mounted on the engine bearers in order to reduce the transmission of the engine vibration to the hull. The mounting flanges should be isolated from the bolts and bearers to prevent abrasive wear of the plastics and enlarging the bolt holes which would aggravate wear and vibration. Suitable designs for avoiding this problem are shown in Figs. 12(a) to (c). An alternative to the above is the flanged plastics girder with the engine fitted on resilient mountings as in Fig. 12(d), this detail, as well as Fig. 12(a), is only used on sailing yachts and small motor cruisers of low horsepower.

Wood bearers are quite common and are often fitted in prototype craft as they prove economical because of alterations often required when re-engining is undertaken.

In larger installations the usual practice is to fit heavy steel or alloy sections well bolted to plywood or timber girders as in Figs. 12(b) and (c). The metal section should be as long as practicable, preferably the full length of the engine compartment and well connected to the bulkheads. The girders should extend at least the full length of the machinery space. Appropriate transverse floors and anti-tripping brackets of plywood or plastics are matted to the hull and preferably bolted to the main girders through metal or plastic angles or wood fillets.

In still larger installations, it is common to build oil fuel tanks into the engine compartment in such a manner to serve as foundations for the steel or alloy engine girders. A typical arrangement is shown in Fig. 4. The girders are through bolted to the heavy sides of the tanks at the appropriate shaft angle, the bolts being matted-over on the tank side. Such a tank structure adds greatly to the strength and rigidity of the hull.

Fig. 12(e) is an all-plastics design which can be used for the larger engine; the section is formed over a heavy density foam core with a well splayed base for stability, and incorporating a steel flat bar in the face.

The outboard/inboard installation where the engine is directly bolted to the transom is becoming very common. The standard practice is to build the transom as a sandwich laminate with a structural core of 1 in. (25 mm.) plywood. The plywood is bedded-in with wet mat or a resin/microballoon mixture. Heavy plywood brackets, matted to the transom, are fitted on either side of the engine and matted and bolted to two short bottom girders which extend from the transom to the next bulkhead.

4 . 5 DECKS AND SUPERSTRUCTURES

4 . 5 . 1—Construction

It is almost standard practice to fit a plastics deck and superstructure, generally moulded as a single unit, in production line craft below 40 ft. in length. In the larger yacht, as in some of the smaller designs, the deck and superstructure are normally custom built to suit each owner and are therefore sometimes of conventional wood construction. The main advantage of the plastics design is that it is one piece and therefore watertight, whereas the main advantage of the wooden deck is that the construction is flexible and can be tailored to suit each boat. The wood design also has a much better appearance than the cold lifeless plastics structure and some discerning owners compromise by fitting a plastics deck with a wooden superstructure.

The structure may be single-skin construction throughout with stiffening section beams or may be a combination of sandwich construction on the horizontal deck surfaces and single-skin on the vertical coamings.

Balsa wood is the most common sandwich core material and is used in many American-designed yachts building in Europe for the American market. The usual procedure is to mould the thin outer face laminate in a female mould in the normal manner and then lay the balsa wood in small blocks or strips, about $\frac{3}{4}$ in. thick (20 mm.), bedded on to a thick resin mixture. Any end grain balsa must first be sealed to prevent it soaking up the resin. The balsa is then sanded flush and the inner face laminate completed. If a teak deck surface is desired, athwartship wood beams, saw-cut to facilitate bedding down, are fitted into the core to take the deck screw fasteners, as in Fig. 13.

Another variation in sandwich construction is the foam core deck in Fig. 14(a) where the foam core strips are matted-in in the same manner as stiffening sections and the core completed by inserting a similar foam strip before laminating the inner face. Fig. 14(b) shows an unusual method of combining two separate mouldings, the outer being the flush deck surface and the inner moulding forming the beams and the smooth finished interior surface. The inner moulding is phased slightly ahead of the deck so that it can be laid on just after the last layer of the deck has been laminated.

Additional material is moulded into the deck structure at stressed corners of deck openings and in way of the mast shrouds. These areas are adequately supported by either bulkheads or beams, particularly at the mast and at the fore end of the deckhouse where the structure has to take the thrust from the shrouds and from heavy seas on the bow quarters. Additional strips of reinforcement are laid along the lower and upper coamings of the deckhouse structure to provide the necessary rigidity and strength continuity, especially where the stiffeners on deck, coaming and deckhouse top, are not continuous. The deck laminate is also increased locally under deck fittings, such as eyeplates, stanchions, etc., and often a metal insert is incorporated.

Often an "anti-skid" pattern is moulded into the deck surface to give a more positive grip than the smooth gel coat finish. The pattern can be very effective in setting off the covering board areas, outlining a simulated king plank and in general introducing appearance to an otherwise smooth

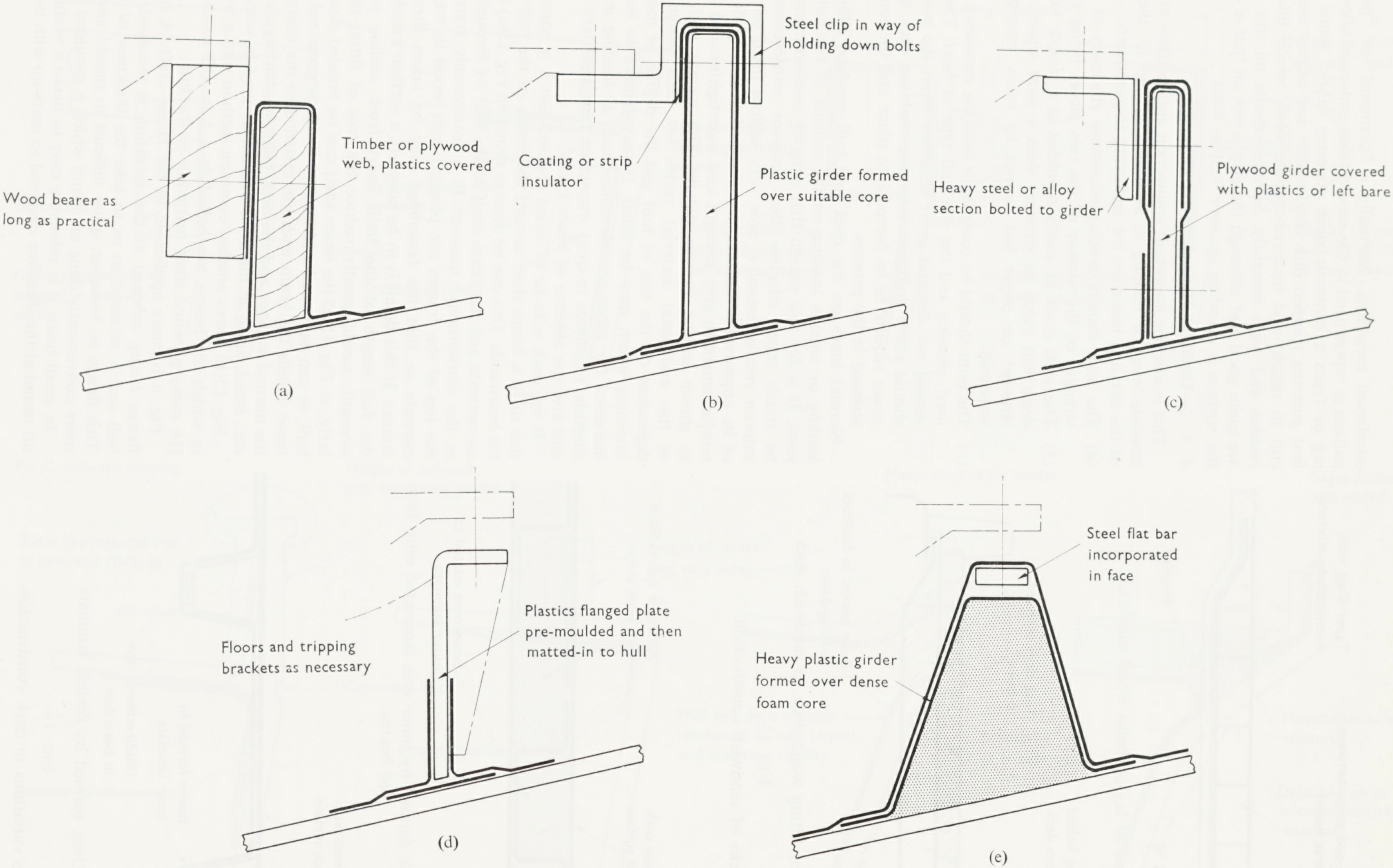
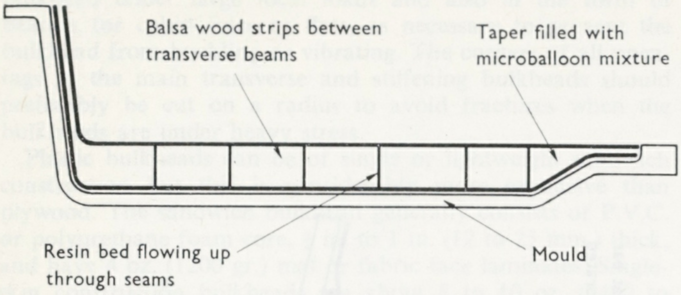
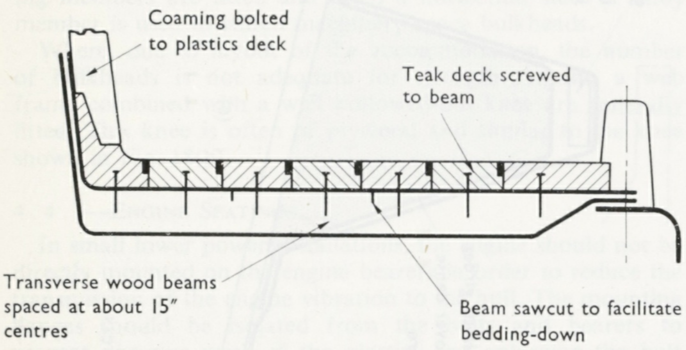


FIG. 12
Typical designs of engine bearers.



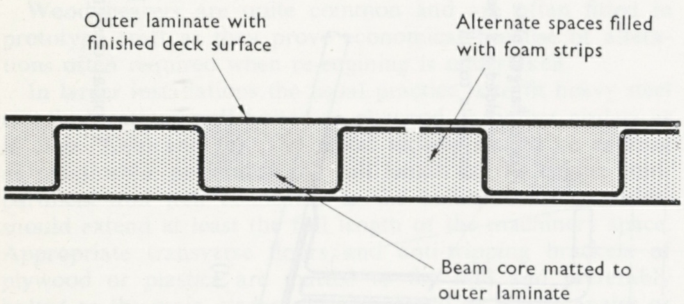
(a) Method of lay-up balsa wood sandwich.



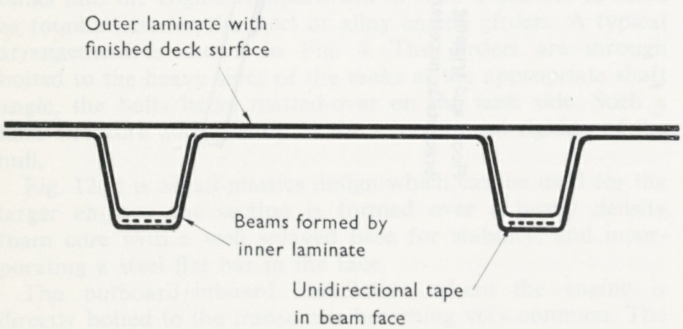
(b) Method of fitting wood deck to sandwich deck.

FIG. 13

Decks of sandwich construction.



(a) Sandwich deck formed by inner skin laminated onto face of beams.



(b) Deck formed by double laminate.

FIG. 14

Two variations in deck construction.

unrelieved moulding. Normally a "diamond" or "pinhead" pattern is reproduced in the mould from embossed alloy sheeting or from a synthetic deck covering fabric. Some of the first patterns trapped dirt very readily and others were difficult to mould and incurred minute voids which filled with residue and were unsightly. Service reports on this surface are quite good and although the grip is not all that is desired, the surface is standing up very well to wear.

4 . 5 . 2—DECK TO HULL CONNECTION

This is another important connection and as there are many possible ways of making it, as shown in Fig. 15, the choice of the detail is based on the following requirements:—

(*a*) The joint should develop maximum efficiency of the full strength of the weaker of the two pieces being joined.

(*b*) The joint must be easily moulded as a simple joint well done will often be stronger than a joint which appears stronger on paper but cannot be easily moulded and inspected.

(*c*) The joint should be compatible with the moulding method used, phasing and the size and type of boat. The design should not demand great accuracy from the mould but should allow sufficient tolerance that the deck and hull never require to be forced into place and the joint overstressed in the process.

Normal loads on the deck and hull produce either shear loading or tension loading on the outside of the hull at the joint. It is for this reason that exterior reinforcement would be much more effective than interior reinforcement, but exterior reinforcement is now very seldom fitted on account of its appearance. Current practice is therefore to fit interior reinforcement to give strength and watertightness to the joint as shown in the various details. The weight and breadth of this additional material varies for each design and is dependent on the size of hull and the degree of mechanical fastening which may be fitted. It may require to be further increased for other considerations such as loading in way of shrouds, resistance to quay impact on bow and stern quarters and also the absence of beam knees.

If the deck is to be of wood construction it is normal practice to fit a beam shelf around the inside of the hull to carry the beam ends. This can be done as shown in Fig. 15(a) by fitting an outwale and the beam shelf to the hull by through bolting in the conventional manner. In a transversely framed hull, the top of the frames are plugged with wood in way of the outwale so that the fastenings may be taken through the frames. If this detail is to be adopted, a vertical knuckle along the full length of the hull, apart from adding additional strength, would greatly reduce the cost of fitting-out. Similarly, as Fig. 15(c) the beam shelf can be bonded direct to the hull, or can be bolted and bonded if this is preferred. In both these details the beam ends are fastened to the beam shelf in the normal manner and the deck planking, carlings, coamings, etc., fitted, again in the same manner as used in wood yachts.

Fig. 15(b) shows another common method with a wood deck in which the beams are bolted to a $\frac{1}{2}$ in. (12 mm.) or $\frac{3}{4}$ in. (16 mm.) plywood shelf matted-in to the hull.

Fig. 4 is more applicable to the larger craft. A horizontal flange facing inboard at the sheerline is moulded with the hull and is recessed so as to take the thickness of the deck. This flange is, however, quite difficult to mould and can be more conveniently done on a hull which is moulded in halves.

In small boats it is relatively easy to make a vertical flange all round at the sheerline designed to mate up with the vertical

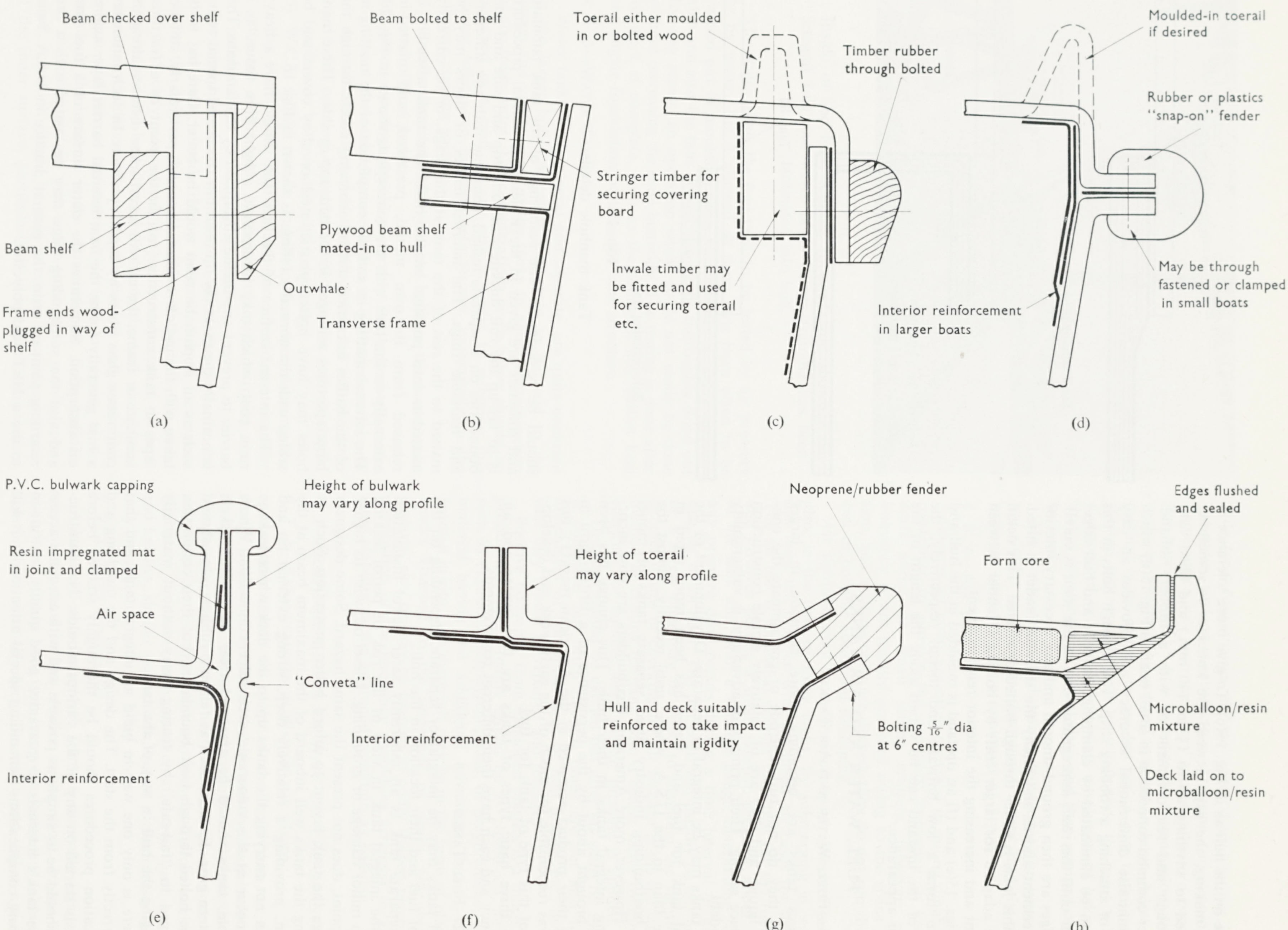


FIG. 15
Typical deck to hull connections.

knuckle on the hull as in Fig. 15(c). To give more tolerance in mould making, the gap between the knuckles is sometimes increased to anything up to $1\frac{1}{2}$ in. (40 mm.) and filled with resin-dough and reinforcement or with a timber ribband. Another common connection in small boats is Fig. 15(d) which does minimise dimensional accuracy and provides an easy means of attaching a rubbing strip. The through bolts in this joint can be eliminated by clamping the deck and hull edges together until the resin impregnated mat in the joint is cured. The edges are then ground smooth and any necessary interior reinforcement added, and finally the snap-on fender is fitted. Often this joint is either through bolted or riveted. This detail can be adapted for larger craft by bolting a fender between the joint and increasing the interior reinforcement.

In Figs. 15(e) and (f) an upstand is moulded in the hull and deck to form a low bulwark and toerail respectively, the height of the upstand can be varied along the length of the hull for appearance.

4.6 FUEL, WATER AND AIR TANKS

4.6.1—Fuel, Water and Air Tanks

Plastics tanks are becoming more common in larger craft as they do not rust, corrode or contaminate the contained liquid, and they are of relatively simple construction and space saving. Their primary disadvantage is vulnerability to accident.

The tanks may be integral or separate. Development of the integral tank for fuel and water has been more rapid in Europe than in the U.S.A. where, until recently, almost the entire boatbuilding industry fitted separate tanks of monel metal. However, more American boatbuilders are now incorporating integral tanks in their hulls. This change has been partly brought about by the persistence of a British firm in having their product accepted in the American market; this firm was responsible for much of the early work and development of this type of tank for fuel.

The three basic types of tanks are fuel, water and air buoyancy and each have their different requirements.

4.6.2—Fuel Tanks

Fuel tanks may be integral or separate, depending on the type of fuel and their location in the hull.

A separate tank is an inherently safer petrol installation than the integral tank in view of the risk of petrol leaking due to minor damage or grounding. Diesel oil, with its higher flash point, does not present the same hazard. Some designers arrange the tank layout to afford better protection. Such as keeping the tanks well inboard of the maximum beam at the section, providing a relatively deep, strong external fin and also in not carrying the tanks up to the deck where leakage may occur at the vulnerable deck edge connection. Where possible, tanks should be arranged clear of shaft brackets and stern gear. Fittings, such as fenders, rubbers, etc., should not be bolted through tank boundaries, but their connection arranged to "fail-safe" by tearing away without materially damaging the hull in way of the tank.

There is only one way to build an integral tank, and that is correctly from the start. The design and the moulding and fabrication procedures should be thoroughly studied before work is started on any of the various moulds. The construction should be as simple as possible and with adequate access to the tank's internals to guarantee good quality matting-in and any complications demanding special attention and skill should be re-designed. The scantlings of the tank laminate and stiffeners are given in the Rules. The normal procedure is to lay-up the full thickness of tank top and side and end boundaries on a separate mould, complete with any stiffeners and manhole rings. The internal baffles or floors may be matted to the rest of the tank at this stage, or be currently matted-in into position on the hull. The tank assembly is released from its own mould, presented, and matted-in externally to the hull with the connections shown in Fig. 20. The internal matting-in and the completion of the matting-in of the baffles is carried out by achieving access through the large apertures arranged in the tank top or sides. These manholes may have bolted plastic, steel or alloy covers and be sealed with the normal jointing, as shown in Fig. 16.

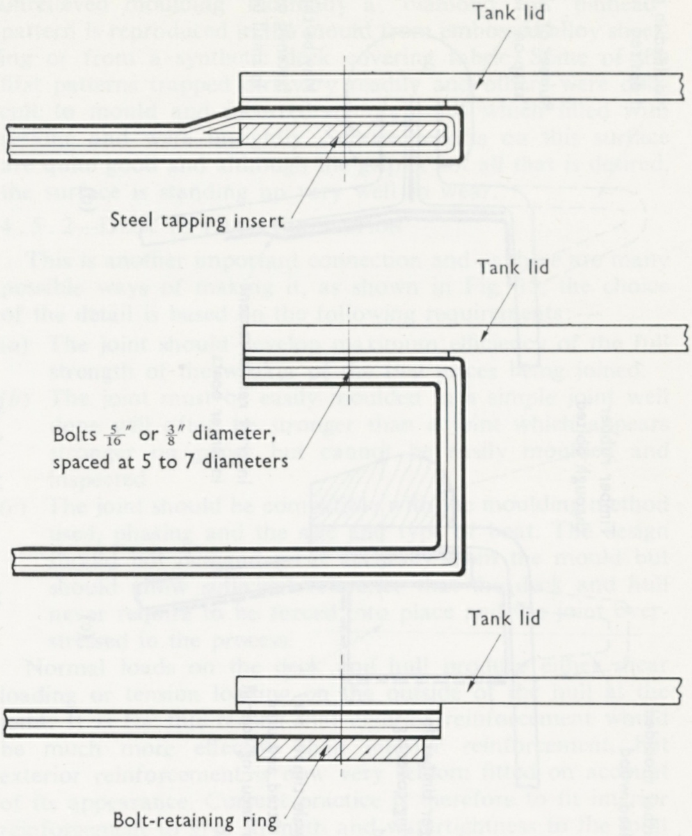


Fig. 16

Tank manhole details.

The internal surface of the fuel tank should have a heavy resin coat, which may incorporate a light fibre tissue, as a barrier to prevent any undue absorption by the laminate. This is normally the main lay-up resin but the use of special resins such as an isophthalic resin would be better. Care has to be taken with the use of roving fabrics and, particularly, stiffener tapes in tank construction for, in the event of a fractured laminate or frame, the oil will travel some distance along the continuous glass fibres by capillary action. In such laminates, a mat generally forms the final internal barrier layer against oil absorption. Sometimes the outer surface of a fuel tank and also the surrounding bilge may be coated with a fire-retarding paint or resin, the general practice in U.S.A. being to use a HET acid polyester.

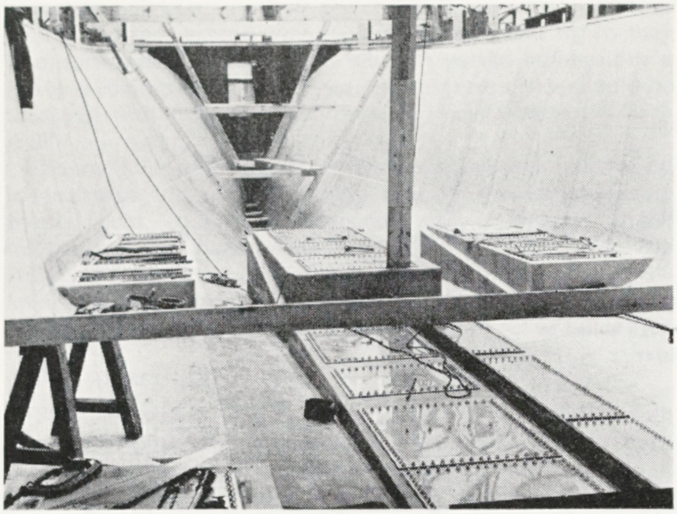


Fig. 17
Inside of the hull, looking forward, and showing the integral fuel and water tanks before framing has been laminated into hull.

The Rules allow the fuel tank to be tested by air pressure or by water. Air is allowed as these tests have often to be carried out in moulding areas where water is highly undesirable and also the tank would have to be thoroughly dried out for any repair, an operation which could not easily be done without increasing the shop humidity. The testing is done after all work is complete, such as the fitting of any ballast keel and engine foundation bolts to the tank structure.

4 . 6 . 3—Water Tanks

Water tanks, both integral and separate, are quite common in the larger hulls. Attention has again to be paid to the design and construction of the tanks to avoid contamination of the fresh water. The considerations given below should be observed in a craft of any size in order to minimise this danger which can be extremely serious, particularly in the offshore cruising yachts:—

(1) A cofferdam should be arranged between the water tanks and oil fuel tanks.

(2) Access openings should preferably not be located at the top of the tank, but are better in a vertical surface to minimise the amount of dirt which can fall into the tank. If the opening must be located in the top, it should preferably be provided with a coaming.

(3) W.C.'s should not be installed on a deck forming the top of the tank and any soil lines should not run over the top of the tank. Neither should pipes carrying non-potable fluids run through the tank.

Troubles have been experienced where the water in the tanks has been tainted. This is believed due to incomplete cure of the resin on the inside surface of the tank. To avoid taint the Society recommends the lay-up resin used in the tank structure and the hull structure forming the tank should not contain any additives other than the catalyst and accelerator, as such additives as antimony oxide tend to migrate to the surface of the laminate in the course of time. Some builders coat the inside surface with a non-toxic resin, usually epoxy based. Attention should be paid to ensure a satisfactory cure of the inner surface, particularly if it has been subject to a little local touching-up, and no water should be put in until the entire surface is satisfactory. A curing and cleaning technique which has been used very successfully on a production line basis in the U.S.A. is to purge the styrene odour from the tanks by injecting live steam for about an hour. A less drastic method is, of course, to use hot air. A new tank should be flushed out several times by which time the taint will have disappeared or, if still remaining, a solution of baking soda that is let stand for several days and then pumped out, may be used.

The construction and testing techniques for water tanks are similar to those for fuel tanks except that the air test pressure need only be a minimum of 3 p.s.i.

4 . 6 . 4—Air Tanks

Integral air tanks are often provided in sailing dinghies and runabouts to provide the necessary buoyancy to keep the craft afloat. In small sailing cruisers, however, positive buoyancy is not usually provided owing to the space limitation, and as these craft are decked, they do not run the same risk of possible accidental swamping.

Air tightness may be determined by air pressure tests and also, in the case of loose tanks, by submerging in water as has been the practice with the metal air cases for lifeboats. The air test is somewhat simpler than that for the fuel tanks and involves putting a small positive pressure, about $\frac{1}{2}$ p.s.i. in the tank and, with a suitable gauge, determining whether any pressure drop occurs during a time interval of about a quarter of an hour. The test cock will usually be left in place or a screw plug may be fitted to maintain the tightness. A simpler means of checking for air leakage is the old standard method of coating the tank with a soapy water solution and checking for bubbles when the air pressure is applied.

The tests should be carried out after all work on the tank is complete, but often integral tanks are cut into for the attachment of additional deck or hull fittings, etc., in which case they should be repaired and retested.

Positive bouyancy from expanded plastics is a much better proposition. These foams should be a "closed-cell' structure such as expanded polystyrene, p.v.c. or polyurethane, the last material having the additional advantage that it can be foamed in situ. The materials may be in the form of planks or blocks attached to the underside of decks, side benches, thwarts, etc., or may be foamed in oddly shaped places. The distribution of the buoyancy should be arranged to give as good stability in the swamped condition as possible.

4 . 7 ARRANGEMENTS IN WAY OF THE MAST

4 . 7 . 1—Arrangements in Way of the Mast

Attention should be paid to the mast area in sailing sloops in view of the varied nature and the magnitude of the loads imposed by the mast and rigging on the plastics hull. The following section deals with the main precautions which are taken to stiffen the yacht and prevent it from working in this area. In yachts with an alternative rig, such as yawls and ketches, similar precautions, but to a much lesser degree, may be required with the after mast.

4 . 7 . 2—Hull Stiffening

A bulkhead or a heavy web frame is normally arranged in way of the mast in the smaller sailing yacht. In the larger boat, it is usual to have twin transverse bulkheads, about 3 ft. apart, which normally form the toilet and locker spaces. A

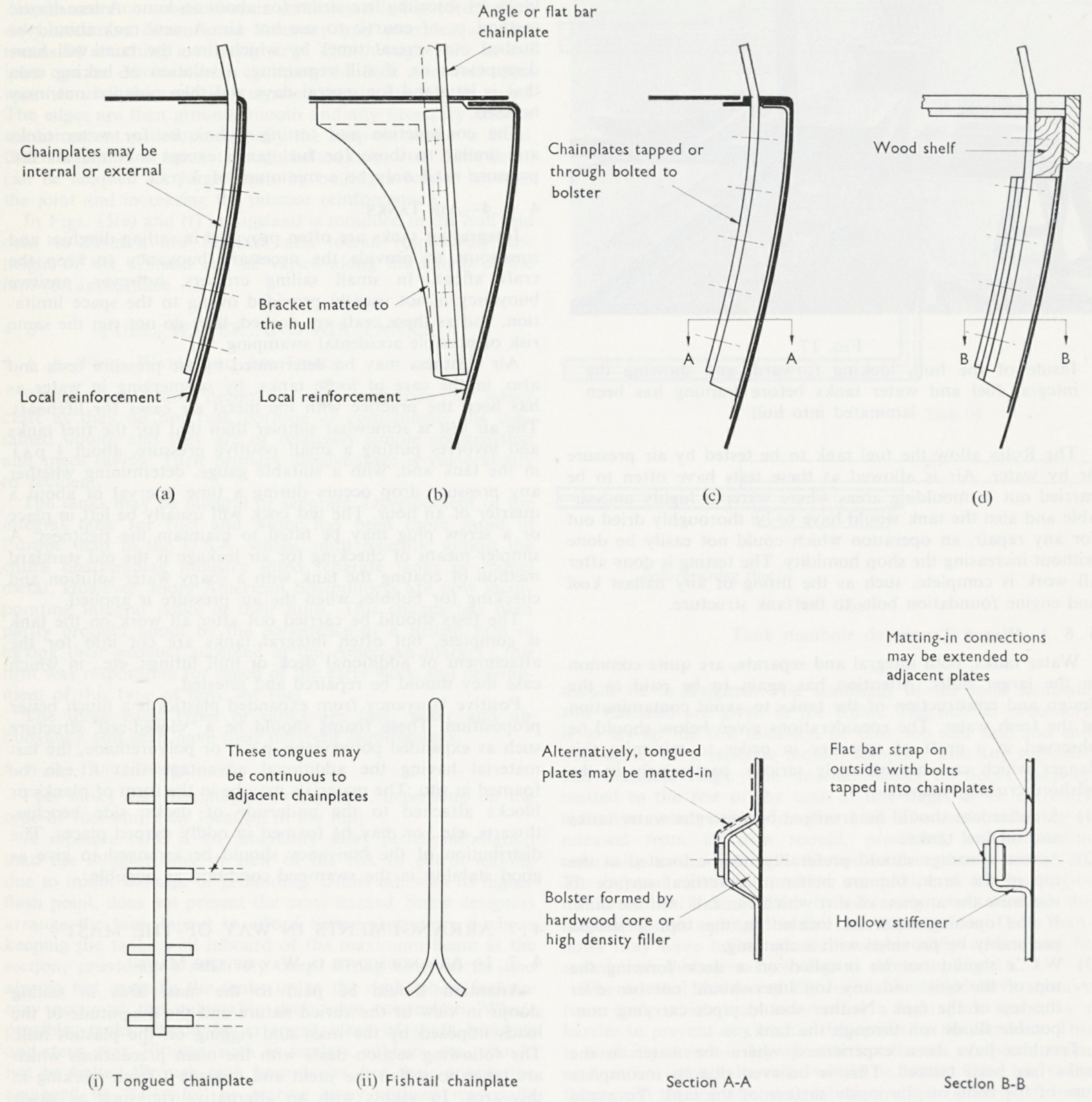


FIG. 18

Attachment of chainplates.

complete bulkhead is much preferable to a partial bulkhead which does not extend over the full breadth and which may also only extend to deck level. Where a partial bulkhead or a heavy web frame is fitted, a heavy transverse beam should be arranged with well bracketed ends carried down to the deck level.

The modulus of the transverse frames in this area is increased to maintain the overall rigidity and in the longitudinally framed boat, equivalent provisions, such as increasing the longitudinal and the laminate thickness may be necessary. The Rules also require the side shell to be increased at gunwale level and this is often incorporated in the matting-in of the chainplates or in the deck edge connection.

4 . 7 . 3—Mast Step on Keel

The mast step is usually formed by a heavy top plate with a centre longitudinal web and several transverse webs in much the same manner as in a steel yacht. This structure may be of plastics or of metal, and is a bearing fit on the keel laminate before being matted-in. The keel reinforcement should maintain its midship width and weight to just beyond the mast step, or the mast bulkheads, and then taper to the stem head. Often in the smaller boat a solid plastics chock is built up on the keel backbone and one builder of the larger boats builds up a similar chock on top of a steel reinforced keel backbone.

4 . 7 . 4—Mast Step on Deck

In boats up to 40 ft. in length, there is a tendency now to step the mast on the deck or the cabin top. This was previously the practice only in the smaller yacht but is now spreading to the larger hull to reduce both initial and running costs as well as give an improved accommodation layout.

The mast step should be located directly over a bulkhead or the transverse strong beam, and the laminate in way should be increased in thickness for an extent beyond the step fitting. In the twin bulkhead arrangement, a heavy longitudinal pad, which may be plastics but is more often of plywood, is normally fitted under the deck between the bulkheads. The width of this pad should not be less than that of the step fitting and it should be checked into the bulkheads as well as adequately bonded to the deck or cabin top. Sometimes the step fitting is designed to span the distance between the twin bulkheads. Where possible, a suitable pillar should be arranged into the bulkhead structure to distribute or transfer the thrust from the mast.

4 . 7 . 5—Chainplates

Fig. 18 shows a number of ways of attaching chainplates. The plates should be ample in size and well fastened to the structure to distribute the load.

Fig. 18(a) is the basic single strap design which may be fitted internally or externally and through bolted. In the case of internal fitting, the bolts should have large heads as, on account of appearance, washers are not normally fitted on the outside of the hull.

Another common method is Fig. 18(b) where the bracket is usually plywood either matted-in or encased in plastics. The plywood bracket is used when a wood deck is fitted as it can be conveniently bolted to the beams, although in some designs a bronze knee is used. Sometimes the chainplates are arranged to pick up a bulkhead and this is, of course, preferable to the knee. The fittings may be single straps or can be double, one being arranged either side of the bracket. Figs. 18(c) and (d) are further methods but are not so common. Figs. 18(i) and (ii) show variants of the single strap chainplate designed to prevent withdrawal and it is essential to ensure the fittings are lying into the hull form before being matted-in.

Where the chainplate and bolts penetrate the deck and hull these should be made watertight with a flexible sealant rather than the rigid resin which may crack under the strain and result in annoying leakage.

4 . 8 CONNECTIONS AND FASTENINGS

4 . 8 . 1—Bonding

The various categories of bonds can be classified as follows: —

(i) Primary bonds are those between successive plies of reinforcement laid and cured at the same time.
(ii) Secondary bonds are those made between tack-free laminate and successive in place lay-up.
(iii) Bonded joints are joints between two previously cured or gelled laminated parts.

The lay-up of the hull, deck, tanks and other assemblies will always be either primary or secondary bonds depending on the size of the moulded unit and the phasing. The matting-in of the various assemblies to the hull should be by secondary bonds and the phasing so arranged that this can be done. The normal lay-up resins are generally designed to remain in a "green" state for 24 to 48 hours in a temperate climate.

The main factor in the design of joints should always be to make the bonding area as large as possible. The joints should be so designed that the resin bond is in shear and therefore lapped joints are preferred. In butt and scarph joints the bond is in tension and the joint should be overlaid with additional reinforcement on one or both sides. Fig. 19 shows typical bonded joints and indicates their comparative degree of stress concentration.

Whatever type of bonded joint is chosen, the areas involved should first be rendered free from release agent, grease, dust, then roughened to expose the glass fibres to ensure a better bond, and the dust removed. The gel coat should be removed in way of any surface to be bonded. Secondary bonds can be done with a general purpose polyester resin, but epoxy resin should normally be used in bonded joints. Polyester is often satisfactory if the cure of the laminates is not too advanced and the area is generous. When polyester resin is used the resin film should be thick enough to prevent difficulties due to undercure, and often a single ply of open texture cloth or mat included in the joint helps to overcome this problem.

4 . 8 . 2—Matting-in Connections of Structural Members

The connection of rigid internal stiffening members, such as bulkheads, webs, girders, tank assemblies, etc., to the relatively flexible shell and deck are important as this connection may create a hard spot which may lead to fractures.

Examples of typical connections for matting-in stiffening members are shown in Figs. 20(a) to (e) ; these diagrams apply to members of single skin or sandwich laminates, plywood or timber materials.

Fig. 20(a) is the basic matting-in connection. The flange thickness and width are significant as the angles shrink during cure and tend to pull the stiffening member through the laminate causing a slight bump. This is particularly noticeable

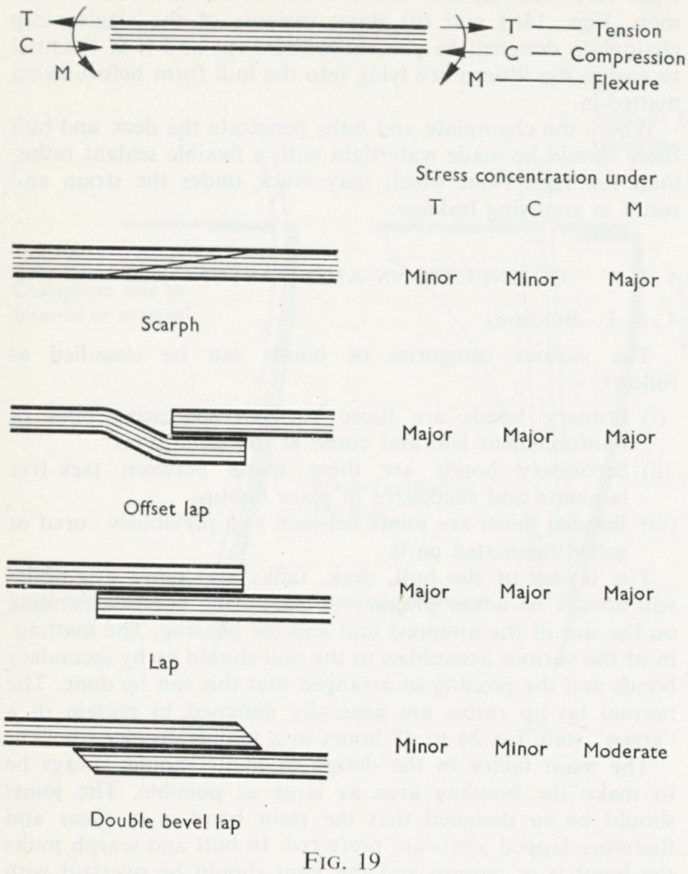


FIG. 19

Typical bonded joints.

when incompressible frame cores are completely encased in the laminate and also when plywood and timber stiffening members are used. The connections are built up from layers of increasing width to reduce the shrinkage effect. In the case of timber stiffening members, the layers should preferably not have the same flange width as there is a tendency for each successive layer, on shrinking, to lift the preceding layer off the member.

This shrinkage effect is the reason for leaving the small gap between the stiffening member and the laminate shown in Figs. 20(a) and (d), which depend on the flexibility of the angles to reduce the hardness and distortion. Another solution is the use of low density core fillets, as shown in Fig. 20(c) ; which also eases the lay-up problem presented by the square corner.

Fig. 20(b) is a typical connection of a member which could be under considerable load, or subject to vibration such as engine girders, mast bulkhead, etc. Continuous reinforcement strips are matted on to the laminate to distribute the load and the member bedded down on wet reinforcement or some other suitable mixture.

With timber members it is good practice to coat the matting-in area with a thin resin prior to fitting to achieve better impregnation of the timber and so improve the bond quality. Another method with plywood is to score the area and break the surface of the top veneer to give better adhesion. However, some builders prefer the member to be mechanically locked as well as bonded and do this by drilling holes along the bonding area and pushing the reinforcement into the holes to form a key as shown in Fig. 20(d).

Fig. 20(e) is where a member is bolted and may be formed by either a single or double angle. A single angle should never be used where the loading on the member will cause the matting-in angle to peel from the laminate. This connection is formed by laying-up a single angle against a suitable template which is then removed and a further angle laid up against the first one.

4 . 8 . 3—METAL FASTENERS

GENERAL.—Laminates can be satisfactorily fastened with bolts, screws or rivets. These fasteners should be of a corrosion resistant metal or should be satisfactorily coated if of steel. They are more common in the small boat, being used where it is more convenient and economic to fasten than to bond, particularly the deck to hull connection, and also as locating devices and clamps, while the resin sets. They are used for securing parts that are intended to be portable in service or for maintenance.

The laminate should generally be increased in weight by about 25 per cent in way of joints and flanges. The fasteners should not be closer spaced than 3 diameters apart and should not be less than 2·5 diameters edge distance in mat laminates and 3 diameters in fabric laminates.

The edges of cut laminates should be sealed by brush coating with resin to prevent water absorption and similarly the fastening holes should be sealed with resin or the compound used in the joints.

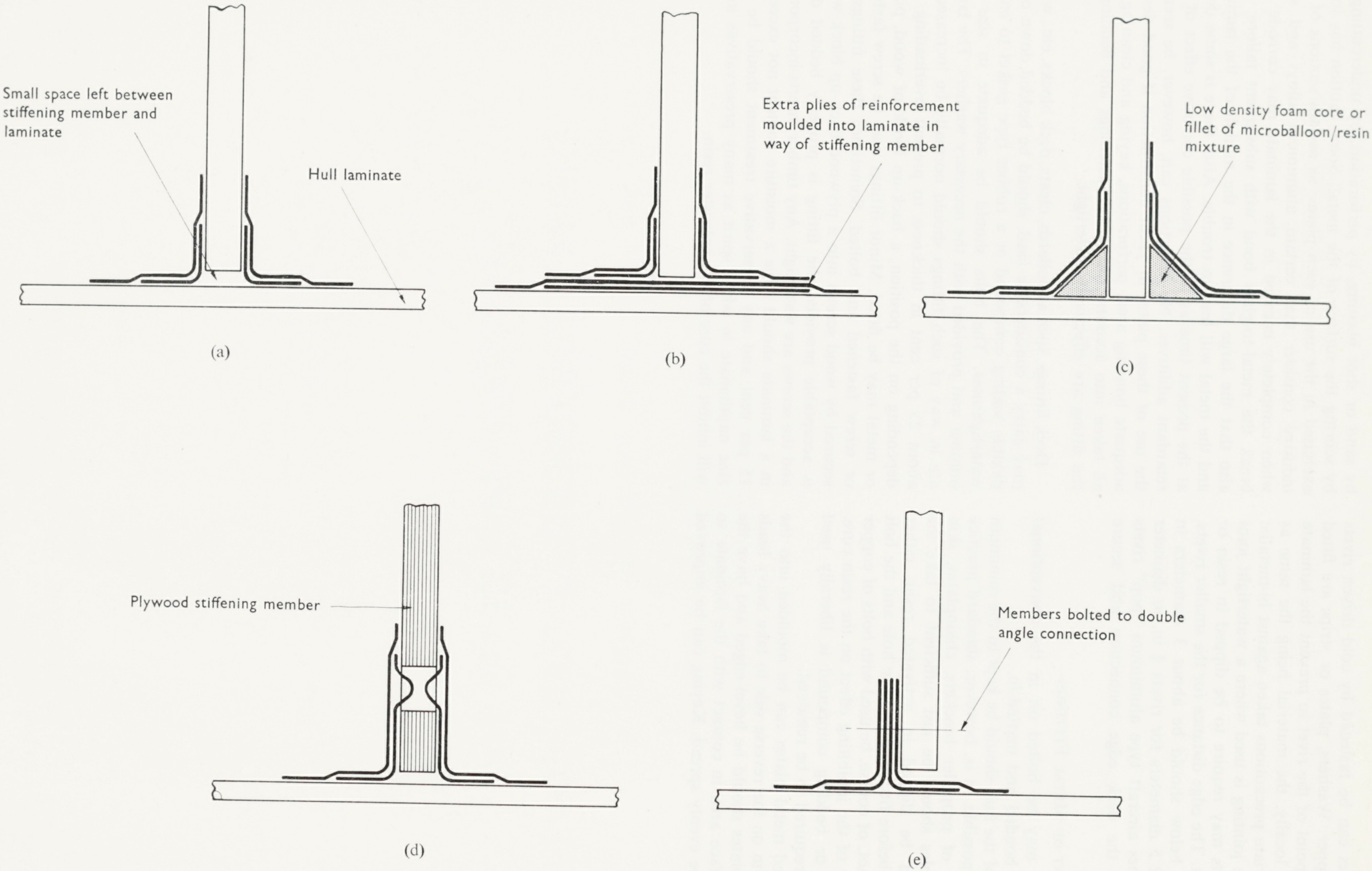
BOLTS.—Bolts should be fitted with generous size washers, not less than 2·25 d outside diameter and 1/16 in. or ·1 d in thickness, where d is the fastening diameter, under both the head and nut to prevent pull out failure of the laminate due to service loading or overtightening of the nut. Countersunk fasteners should have countersunk washers or lips on the head. Bolts, however, should be avoided in laminates less than 4 oz. in weight. As a rough guide the bolt diameter should be approximately equal to the laminate thickness.

A twist drill close to the size of the bolt should be used to prevent possible working and enlargement of the hole. Lock washers are recommended where any substantial vibration is involved. Also where nuts become inaccessible after assembly of the boat they should be prick punched or peened to prevent backing off.

SCREWS.—Self-tapping screw fasteners and tapped Whitworth machine bolts can be used for the connection of lightly loaded items of relatively minor importance where a better type of connection cannot be employed, such as where the opposite side is inaccessible.

As the screw needs sufficient thickness penetration for dependable holding power, it should not normally be used in laminates less than 4 oz. in weight unless screwed into a metal tapping strip or equivalent. A pilot drill the exact size of the screw barrel should be used and the fastener carefully turned home. The screw will self-tap iteself pretty well into the material although the preparatory tapping procedure used in metals is better. The fastener may be set in epoxy to improve the grip.

Screw fasteners should always have their long axis perpendicular to the reinforcement plies and are never screwed into the end of a laminate. Because of the hard incompressible nature of this material, wood screws cannot be fitted properly and are never used.



The sketches are diagrammatic only and do not indicate the number of layers to be used

FIG. 20
Typical matting-in connections of structural members.

RIVETS.—Laminates can be fastened by cold driven rivets of steel, alloy or copper. Washers, plates or strips are fitted under the head and point of the rivet to prevent the laminate from being crushed locally, the material being the same as the rivets or appropriate precautions taken against bi-metallic corrosion. A suitable jointing is used where a watertight joint is required, the rivets may require to be dipped in resin or other suitable sealant. The edge distance for the smaller rivets, $\frac{1}{4}$ in. diameter and below should be about 3 diameters in mat laminates and 2·5 diameters for rivets $\frac{3}{8}$ in. in diameter and above. Sometimes aircraft type aluminium "pop" rivets are used to fasten the deck edge connection and secure internal linings.

4 . 8 . 4—ATTACHMENT OF METAL FITTINGS

The metal fitting may be bolted on in the conventional manner or may be bonded and matted-in.

Through bolting of the hull should be kept to the minimum and avoided where possible. It is, however, standard practice for the attachment of propeller brackets, chainplates, skin fittings, etc. The holes should be just sufficient to take the bolts which should be dipped in activated resin, either polyester or epoxy, before fitting to seal the hole and the bolt head. Polyester cannot, of course, be used with bolts of copper or its alloys in view of the inhibiting effect on the resin cure. A suitable sealant or bedding compound is liberally used where the fitting is required to be removed.

Drilled and tapped metal plates can be moulded into the laminate or matted-in on the reverse side to take heavy loads from fittings. The plates should be bevel-edged and have the largest possible surface area in contact with the laminate so that the load can be evenly spread. Keying can be improved by sand or shot blasting, by perforation, by undercutting or by scoring the surface of the metal. Some caution has to be exercised in the use of such plates, as certain sections of the industry consider that certain aluminium alloys and steel when completely encased in the laminate may corrode and break the metal-to-glass bond with subsequent failure, and also that the large difference in the moduli of the laminate and the metal will lead to trouble. Also, there is some doubt at the present time as to the possible corrosive effect of fire retardant additives. No objections can, however, be seen to the use of these plates in yacht construction if good design, adequate bonding area, perforations, keying and cleaning are all taken into account and if the fastenings and bedding of the fitting are effectively watertight.

Deck fittings, such as bollards, cleats, deck blocks, etc., which may carry a considerable load, should be bedded down on a flexible sealing compound or a rubber type gasket to ensure watertightness. The bolts should be adequate in size and number and provided with the necessary washers. The laminate in way of such fittings should normally be increased by about 25 per cent in thickness to prevent overloading, and depending on the position, a back-up block of wood, plastic or metal may be fitted. Minor fittings may be screw fastened or screw fastened and bolted. Sometimes these fittings are secured by wood screws into a plywood back-up block which is acceptable provided the fitting is properly bedded down and the screws are watertight. Any timber inserts incorporated in a laminate should have a moisture content not exceeding 15 per cent and any preservative treatment should be with zinc napthanate in white spirit as many preservatives in use will inhibit the cure of the polyester resin.

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Session 1963 - 64

Paper No. 4

Discussion

on

Mr. W. L. Hobbs' and Mr. A. McInnes's Paper

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The Authors are very grateful to all who have contributed to the discussion. For ease and convenience in reference, they have rearranged some of the contributions under the various sections and hope that this will meet with the contributors' approval.

GENERAL

MR. J. B. DAVIES (Headquarters)

During the discussion on Parts I and II of this paper, which were read last Session, I remarked that we were all looking forward to seeing the next parts and, now that we have them, I am sure everyone will agree that they have come up to our expectations.

Mr. Hobbs has given us a most valuable description of the moulding of a hull and has brought out many points requiring particular attention of the Surveyor. There are only a few points I wish to raise.

It has been my experience in discussions with moulders that their costing system is far better organised than it is with the traditional boat builder. The latter generally repairs yachts and provides lay-up facilities as well as building, and at the end of the year will know whether his bank balance (or his overdraft) is better or worse than it was at the beginning, but rarely knows whether this is due to a profit or loss on building, repairing or winter storage. Consequently, his new construction price against which the plastic moulder is competing, may or may not be realistic. Obviously, the moulder's cost per boat will be effected by the procedure he adopts for writing off the fairly high cost of a mould. I wonder if Mr. Hobbs has any information on the normal procedure in this respect.

When discussing future possibilities, Mr. Hobbs suggests "composite" metal and plastic construction as the next stage. Personally, I think we may well find that sandwich construction will satisfy our needs for some time ahead.

I have no comments on Mr. McInnes's part of the paper other than to say how useful I am sure this will be. I understand we still have another paper to come from Mr. McInnes so I can conclude, as I did last time, by saying how much we are looking forward to it.

AUTHOR (W. L. Hobbs)

I agree with Mr. Davies that the costing system of reinforced plastics moulders is, generally speaking, better organised than that of traditional boat builders despite the fact that two plastics boat builders in the Southern Region have had to wind up in recent months. It should, however, be remembered that whereas the traditional boat builder is dealing with several trades and several materials, one of which latter is very wasteful in conversion, the reinforced plastics moulder has only two main materials, both of which are priced on weight, and which with proper factory organisation call for little waste allowance. Accurate costing of plastics mouldings is therefore a relatively straightforward matter.

The mould cost per boat will depend on the number of craft to be moulded, and in the runabout field this may amount up to a hundred craft while in the larger yacht field, one firm reckons that, provided six craft are produced from one mould, they are on a good wicket for subsequent craft produced from the same mould.

Regarding sandwich construction, foamed P.V.C. gives a very strong form of construction with G.R.P. skins for craft operating under normal temperature conditions but becomes unstable at 60° C. and is, therefore, unsuitable for deck construction for craft destined for use in tropical waters, and I would think also rather risky if used in the topside shell laminate of such craft. Foamed polyurethane is a rather brittle material and disintegrates easily when subjected to vibration, and balsa wood rots quickly if water enters through punctures in the glass fibre skins. Perhaps when foams without these faults are developed we will be able to judge this form of construction better. I still think composite metal and plastics construction will prove the cheapest and most reliable form of construction when craft over about 100 ft. to 150 ft. are developed.

3 . 2 . 2—WOOD MOULDS

MR. O. M. CLEMMETSEN (Headquarters)

In the construction of wood moulds the inner skin is stated to be stapled to the outer skin and these staples are subsequently removed. Assuming these are normal staples it would appear that less damage to the mould surface would be incurred by punching them in and filling and I would like to know if the staples are always removed. Possibly the mould has a bearing on this question and perhaps Mr. Hobbs could give some information on this point.

AUTHOR (W. L. Hobbs)

Mr. Clemmetsen's suggestion for punching in the staples of the inner skin of wood moulds has been tried but found to cause more damage to the inner skin than withdrawing the staples and filling the holes.

AUTHOR (A. McInnes)

The staples are the normal type and as each skin is only 2/16 in.–3/16 in. thick, the ends protrude through the outer skin and it is a relatively simple matter to push them back through again. The number of staples used can amount to three or more per square inch in areas where there is considerable curvature in the hull.

3 . 3 . 2—PREPARATION OF THE RESINS

MR. A. LINDQUIST (Abo)

One of the most common faults in plastic boats after a few years of service is probably the appearance of hair cracks in the surface at stressed parts of the structure. These cracks have appeared also where the gel coat has been reinforced by 4½ oz. (160 gr.) cloth. One of the reasons for the appearance of the cracks can, of course, be that the gel coat has been too hard and not flexible enough. Another disadvantage of a polyester gel coat is the non-durability of the colours. This difficulty can partly be overcome by painting the structure but at least in this country the builders prefer to use colour pigments in the gel coat without any painting afterwards. This all ends up in a question: is there a material better than the polyester which could be used, would epoxy give

an advantage if used in the gel coat where the higher price would not matter?

Mr. O. M. Clemmetsen (Headquarters)

What sort of paints are used for painting plastic hulls and what are the disadvantages of pigmenting gel coats? When the gel coats are pigmented are the colours lasting?

Author (W. L. Hobbs)

I feel the above-mentioned hair cracks in the gel coat could have been caused through undercure or through the use of a resin having insufficient flexibility, or perhaps one containing too high a proportion of fillers, or fillers of the wrong type.

The use of epoxy resin for the gel coat, with its superior abrasion resistance compared with polyester resin appears at first sight attractive, but one firm that tried this was unable to obtain proper cure with the subsequent polyester resin/glass laminates and the scheme had to be abandoned. I understand epoxy resin will adhere and cure when applied to a cured polyester laminate but that when the procedure is reversed, the epoxy poisons the polyester. Perhaps in this case the wrong type of epoxy was used and I will be interested to hear if any colleagues know of successful applications of this system.

For painting the hulls of plastics craft, polyurethane coatings have proved very effective provided the surface to be painted is properly prepared. The materials now available for pigmenting the resins are better than those used in the early days and the yellowing previously experienced with white hulls appears to have been overcome. However, some shades are inclined to bleach, especially the greys. Some patrol craft recently supplied for service in tropical waters changed from grey to white in a few months.

Author (A. McInnes)

I agree that the hair cracking of the gel would possibly be due to too brittle a gel coat and one which was possibly based on the brittle lay-up resin rather than a gel coat prepared by the manufacturer to which a certain amount of flexibliser has been added. These cracks often occur at corners of thin laminates, mainly in the deck and cockpit areas, which are deficient in stiffness either in the completed condition or whilst being fitted out.

Mr. Clemmetsen's query on painting will be further dealt with in Part 6.

3 . 3 . 3—Application of the Resins

Mr. O. M. Clemmetsen (Headquarters)

I notice that Mr. Hobbs has a very poor opinion of surfacing tissue, but nevertheless this appears to be one of the standard ways of obtaining greater thickness in the gel coat. Does the use of scrim cloth also give a greater gel coat thickness or is a cloth preferred on account of its greater resistance once the gel coat has been abraded?

Mr. A. Lindquist (Abo)

One manufacturer here has tried to use spray equipment for the application of both resins and glass fibre. The results as regards tensile strengths have not been satisfactory. It would be very interesting to know what experience the Authors have of this type of application and the properties of the finished material. This type of application when successfully applied will give a very high production rate.

Mr. J. B. Davies (Headquarters)

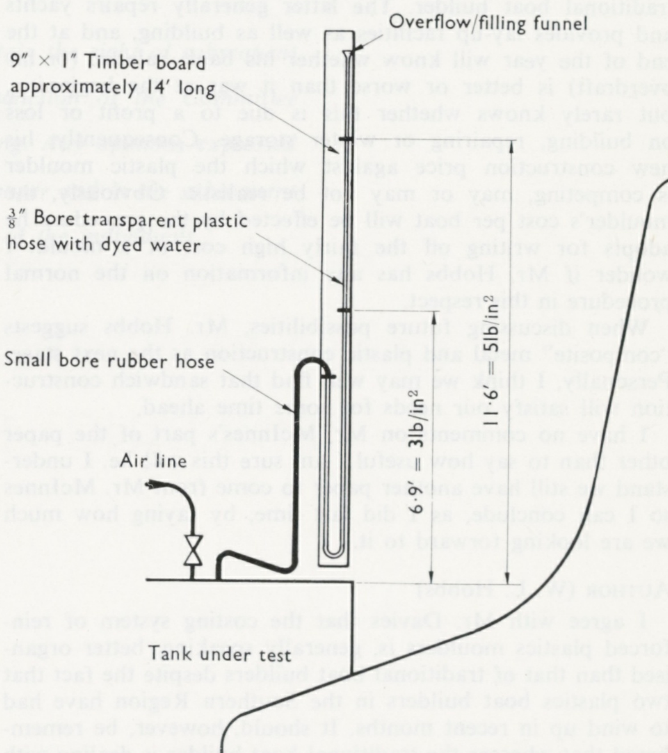
Mr. Hobbs has only dealt with hand lay-up and I would be interested to know his opinion of whether spray guns may come back in favour. Labour is getting more expensive all the time and spray lay-up would be attractive provided the difficulties encountered when the equipment was first introduced can be overcome.

Author (W. L. Hobbs)

Spray lay-up technique certainly cuts labour costs, but I feel this technique is mainly suited for canoe shaped hulls, i.e. craft without reverse curvature at the bilge or deep hollow fins. Rotating moulds are essential for this method and it calls for highly skilled operators. It has proved satisfactory in lifeboat production but it is usual, I believe, to incorporate hand laid-up mats in the areas of high stress, e.g. gunwales and backbones, so it would appear that weight for weight, the hand lay-up technique produces a stronger laminate.

It is possible to obtain a gel coat of sufficient thickness without the use of surface tissue by applying two coats with rollers or brush and I favour a woven cloth of around 6 oz. weight for its greater protection of the subsequent mat layers in the event of the gel coat receiving abrasion damage.

Layout of air pressure test rig.



The above diagram shows a simpler test rig to that shown in Fig. 3 on page 6 of the paper.

3 . 3 . 6—Hull Inspection on Release from Mould

Mr. A. Lindquist (Abo)

Is there any common practice regarding mechanical tests made from material cut from time to time to check the mechanical properties by taking tensile tests? The problem

is that offcuts are never available from the most important parts and we only hope that the properties of the shell are as good as those of the deckhouse from which the tests have been cut.

AUTHOR (W. L. Hobbs)

It is usually possible to obtain samples of the actual hull laminate for test purposes when cut-outs are made for fitting seacocks, portlights, etc.

AUTHOR (A. McInnes)

Mechanical tests are seldom taken as few boatyards have the necessary equipment. Some material manufacturers, however, will carry out the tests on their customers' laminates but this is normally restricted to investigations on laminates where something has gone wrong, otherwise their laboratories would be choked up with routine testing. To have this work done by a commercial laboratory would probably cost more than the survey fee.

3 . 4 . 2—FITTING THE BALLAST KEEL

MR. O. M. CLEMMETSEN (Headquarters)

When ballast keels are completely enclosed in plastic, are keel bolts still fitted or is the keel supported only by the shell? In the latter circumstances is the bottom shell locally increased in way of the ballast keel?

MR. A. LINDQUIST (Abo)

After five years' experience of a small sailing yacht with an external ballast keel I feel that the only correct way to fit the keel is to have it inside the glass fibre. The distribution of stress from the keel, which, above the level of the floors anyhow, has to be taken by the shell and frames, must certainly be better when the weight is carried by the shell directly. The other type of construction with an external keel merely distributes the weight of the keel through the keel bolts to the same structure. The only real danger would be caused by a hard grounding on a rock at high speed cutting the glass fibre underneath the lead, so I think that rather than take the risk of losing the expensive piece of lead I would fit some extra keel bolts to the built-in lead keel. For a racing yacht the internal keel is definitely an advantage as the work to grind the bottom paint smooth in way of an external keel is more than the work spent on all the rest of the bottom.

AUTHOR (W. L. Hobbs)

Encasing the ballast within the plastics hull is becoming common practice. The shell moulding is substantially thickened in way and woven fabrics are sometimes interleaved between the mat layers to give additional strength in the keel area. Some builders fill the hollow fins with lead shot or lead powder but this works out more expensive than cast lead and I feel it could prove rather dangerous should the underside of the shell become holed during grounding. If the amount of ballast is considerable, additional strength in the fin area can be obtained by fitting the lead in a number of cast pieces between the plastics transverse floor members, this method makes for ease of handling.

AUTHOR (A. McInnes)

It is not common practice to fit keel bolts when an internal ballast keel is fitted and as Mr. Lindquist also wants to do this, I feel he need have no fear as it would need a tremendous impact to damage the keel to the degree that the lead ballast would fall out. This was particularly borne out by the case quoted by Mr. Hobbs in the last paragraph in Section 3 . 4 . 2.

There is the problem in the racing boat that an external ballast keel must be fitted if the maximum ballast/displacement ratio is to be obtained and the lowest ballast centre is to be achieved.

Regarding the scantlings in way of an internal ballast keel, the hull laminate is retained at fin weight right round the keel and three or more layers of 2 oz. (600 grs.) mat are carried over the top of the ballast to tie the sides of the yacht together, as is shown in Section 4 . 4 . 1.

3 . 4 . 3—ATTACHMENT OF THE CHAINPLATES

MR. A. LINDQUIST (Abo)

The approximate breaking strength of the shrouds (ex. direct lower shrouds) in a sailing yacht is usually taken as twice the weight of the ballast. When this figure is compared with the amount of glass fibre needed to carry this load we have found one of the big advantages with glass fibre. In an ordinary single skin wooden yacht this load has to be carried by the frames, in case cross ties are not fitted, but in a plastic yacht the unilateral strength helps a lot.

AUTHOR (W. L. Hobbs)

The point made regarding the chainplate anchorage is borne out by recent tests carried out on stainless steel chainplates bolted to steel tapping plates bonded into the hull. The chainplate failed at 1·5 times the displacement of the yacht but the plastic matting-in connections remained undamaged.

3 . 4 . 4—INTERNAL JOINERWORK

MR. A. LINDQUIST (Abo)

Even if the internal joinerwork does not form part of the classification I believe we have to be very careful with these items in the bigger boats as there is always a risk that the flexibility is not taken into account when the hull of a plastic yacht is fitted-out by a boatbuilder used to conventional building. Any accommodation fitted without due regard to the flexibility of the hull will soon result in broken woodwork and would also cast a shadow on the strength of the hull, at least in the owner's opinion. I feel that we have to advise the builders in this respect and it would probably also be a good idea for the builders to submit drawings for consideration of the intended system of fitting the internal joinerwork.

AUTHOR (W. L. Hobbs)

Where locker fronts are of wood, it is quite reasonable, and indeed is now normal practice to bond them to the hull and make them into useful strength members. Plastic boundaries are often moulded to the hull to take the cabin sole and platform bearers when wood soles and platforms are to be fitted.

4 . 2 . 2—SINGLE-SKIN CONSTRUCTION

MR. A. LINDQUIST (Abo)

It would be very interesting to hear the Authors' opinions regarding the difference in weight between longitudinal and transverse framing. I believe some weight can be saved by

using longitudinal framing utilising the transverse accommodation bulkheads with their normal spacing of about 7 ft. as transverses. It would also be very interesting to know the average weight compared with an ordinary single-skin wooden construction, I presume calculations were made in connection with the proposed rules for the I.Y.R.U. 5,5 metre Class.

Mr. O. M. Clemmetsen (Headquarters)

On the construction side of the paper, I note that internal structure in longitudinally framed boats is also used to replace longitudinal frames. May this internal structure be permanent wooden furniture or is it only such items as fuel tanks and engine girders that are referred to?

Author (W. L. Hobbs)

Where transverse bulkheads, either of plywood or plastics foam sandwich construction are bonded to the hull, it is normal practice to omit the transverse frames in way, so it would appear there is no weight saving in adopting longitudinal framing. Compared with single-skin wood construction, I am of the opinion G.R.P. hulls show a saving in weight in craft over about 30 ft. in length.

In the International 5,5 metre Class plastics construction is not yet allowed but judging by the ballast/displacement ratio figure we were able to obtain from a plastics One-Design type built to fit this rating rule, it would appear there was little, if any, saving in the weight in the hull. I believe the designer had in mind the possibility of plastics being accepted if it were proved that plastics hulls suitable for the rating rule were of equal weight to hulls of wood and calculated his shell and framing weights to this end.

Author (A. McInnes)

It is common practice to include wood furniture such as bunk bottoms, shelves, cupboard tops, etc., into the longitudinal framing system provided these items are structurally matted-in.

4 . 3 . 2—Selection of Laminates

Mr. A. Lindquist (Abo)

The builders in this country all use laminates built up of mats and woven rovings. This is mainly because they consider that the impact resistance is of primary importance for small craft, especially dinghies and fast small outboard motorboats. It would be helpful to know the corrections applied to rule weights when substituting mats with different types of roving.

Author (A. McInnes)

On re-reading the text, I appear to have been "harder" on the woven roving fabrics than I intended. Here, of course, I may have been influenced by the British practice. These fabrics will become more and more important as the size of the craft increases, and in the smaller boat where a "frameless" hull is desired. When a woven roving/mat laminate is used in place of the rule all-mat laminate, the correction to the rule weight is generally made on the basis of equivalent flexural strength.

4 . 4 . 3—Framing and Stiffening Sections

Mr. O. M. Clemmetsen (Headquarters)

I should like to ask whether there is any danger of wood cores for framing being deleteriously affected due to being completely enclosed in plastic.

Author (W. L. Hobbs)

Regarding wood cores for framing, providing the surrounding plastics laminate remains watertight, the cores are not deleteriously affected but wood coring is seldom used for frames nowadays.

4 . 5 . 1—Construction

Mr. J. Guiton (Headquarters)

In the section dealing with decks and superstructures Mr. McInnes writes: "In the larger yacht, as in some of the smaller designs, the deck and superstructure are normally custom built to suit each owner and therefore sometimes of conventional wood construction. The main advantage of the plastic design is that it is one piece and therefore watertight, whereas the main advantage of the wooden deck is that the construction is flexible and can be tailored to suit each boat. The wood design also has a much better appearance than the cold lifeless plastics structure and some discerning owners compromise by fitting a plastics deck with a wooden superstructure".

Are individuality and better appearance the only reasons for retaining wood decks? Have the Authors any information about the habitability of plastic boats?

In order to avoid extreme temperature changes a steel boat is usually lined in way of accommodation, whereas a small wood boat is not, owing to the superior insulation properties of wood. Similarly, condensation is not considered a problem in wooden hulls, but cork-spraying or lining must be used in a steel hull. How does plastic compare with these traditional materials?

Author (W. L. Hobbs)

Mr. Guiton mentions condensation and a single-skin reinforced plastics deck is subject to this unless sprayed on the under surface with anti-condensation material. Sandwich construction gives less trouble in this respect, but in passing it should be mentioned that wood decks are not immune from condensation and plywood can be troublesome in this respect.

4 . 7 . 3—Mast Step on Keel

Mr. A. Lindquist (Abo)

Many of the mast steps used in plastic yachts seem to be inherited from the wooden boats where the mast is usually stepped close to the joint between the stem and the keel where the complicated mast step is mainly needed to compensate for the lack of strength in way of the joint and across the planking. A chock on the reinforced keel is probably a nice and efficient solution.

Author (W. L. Hobbs)

Although the mast steps fitted to plastics craft resemble those fitted to wood yachts, they have not inherited the hull leaks that often spring from this locality in the wood hull,

4.7.5—Chainplates

Mr. A. Lindquist (Abo)

Have any tests been made to determine the efficiency of the joints between the plastic and galvanised steel or brass? It would be very useful to have some rough figure for allowable shear stress. One very simple and probably efficient type of chainplate used by, I believe, a Dutch designer consists of a bent round bar with both ends bolted to the strengthened deck.

Author (W. L. Hobbs)

Apart from the chainplate test already mentioned, I know of no way to determine the strength of metal to plastics joints but it should soon be possible to produce figures based upon area to give useful guidance in this matter.

Author (A. McInnes)

Although tests have been made to test the efficiency of the bond between polyester and epoxy with numerous metals, the results have been specific to the particular joint and surface finish and are not available in a usable form for general application. The chainplate mentioned is a very simple design which is becoming popular, passing through the deck in way of the heavy matting-in connection with the hull and fitted with large diameter washers or a washer plate on the underside.

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DEVELOPMENTS IN CARBON STEELS

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DEVELOPMENTS IN CARBON STEELS

By R. E. Lismer

Fundamental metallurgical research in the last few years has revealed that there is a great potential for the improvement of plain or carbon steels. These materials are produced in great tonnages and in order to meet competition from alternatives such as reinforced concrete, plastics and non-ferrous metals, new techniques of metallurgical study have been used to examine more critically their fundamental properties. Computer study has been applied to a very large volume of quantitative information obtained and one of the features revealed is the importance of very small quantities of alloying elements which can be combined with iron, in particular carbon and nitrogen.

These studies really started with the efforts of steelmakers to produce ship plate steel and constructional steels of defined and repeatable quality. The elements studied include carbon, manganese, silicon, aluminium, niobium, vanadium, titanium and nitrogen. The process variables, besides the steel melting, include cross rolling, rolling finishing temperatures, reductions and plate thicknesses, plate cooling rate, cold work and normalisation treatments.

An important metallographic characteristic was the ferrite grain size and in these new steels the grains were so very fine that methods of lineal analysis were developed more selectively to assess the diameters. Electron microscopes were needed to study the very fine form and dispersions of the precipitation hardening constituents, the carbides and nitrides. The discussion which follows concerns those steels developed for low temperature service in which the Society has already defined its requirements as good weldability, ease of formability, and improved notch ductility.

The Society has played a major role in the immense amount of work dealing with the problem of brittle fracture, the results of which led to the formation of related specifications from standards organisations and Classification Societies. The British Standard Specification 2762 (1956) defines the mechanical properties of four grades, referred to as ND I to ND IV. The Society details the requirements of five grades A to E, of which grades D and E correspond approximately with ND II and ND IV, respectively. The differences between the grade requirements are intended to increase notch ductility progressively from ND I to ND IV, and whilst there are many criteria for assessing this property, the most common is the transition curve given between impact energy and testing temperature for the Charpy V-notch impact test.

For the purpose of ship plate manufacture, the results of investigation indicated that the transition curve can be improved simply and economically by one or more of the following procedures:—

(1) Reduce the carbon content.
(2) Reduce the phosphorus content.
(3) Reduce the nitrogen content.
(4) Increase the manganese content.
(5) Do not use rimming steels.
(6) Reduce austenitic grain size by adding controlled amounts of aluminium to the molten steel.
(7) Adjust finishing temperatures at the rolling mill to ensure a fine ferritic grain structure.
(8) Normalise the rolled plates.

A typical ordinary mild steel ship plate might have a composition of about 0·20 per cent carbon, 0·5 to 0·6 per cent manganese. It is usually of semi-killed or killed quality, in the "as rolled" condition. The transition range is above ambient temperature and between +30 and +50° C.

The impact energy values of graded steels ND I to ND IV are specified as follows:—

ND I — 20 ft./lb. at 0° C.

ND II — 20 ft./lb. at −15° C.

ND III — 20 ft./lb. at −30° C.

ND IV — 20 ft./lb. at −40° C.

The manner of obtaining these properties is achieved by increasing the manganese content along with a decrease of carbon content to maintain the tensile range of 28 to 32 tons/sq. in.; by grain control using aluminium additions or other steel-making practices; controlled rolling and finishing temperature; normalising treatment.

The requirements of analysis and works procedure for producing the various plate qualities are very broad and much is left to the option of the steelmakers. A typical procedure would be as follows:—

ND I is a 1 per cent manganese steel with 0·15 per cent carbon content, killed or semi-killed and used in the "as rolled" condition. The movement of the transition temperature is about 20 to 30° C. below that of ordinary mild steel. ND II is more or less ND I in the normalised condition. Such treatment refines the ferrite grain structure, slightly increases the yield strength, slightly lowers the ultimate strength, with a move of transition temperature to about 15° C. below that of ND I.

ND III has a manganese content increased to about 1·4 per cent, balanced by the carbon being slightly lowered, silicon killed and used in the normalised condition. Transition temperature range is down to about −30° C. or −40° C.

With ND IV, the carbon content is slightly increased to give a tensile strength value towards the upper limit of the specified range, with the impact properties enhanced by grain control, controlled plate rolling and normalising temperature. The transition temperature is about −50° C. and it is feasible to obtain such results by the addition of up to 1 per cent alloying element other than manganese. In fact 1 per cent alloy will decrease the transition range down to −60° C.; 2 per cent alloy down to −75° C.; 3 per cent alloy down to −100° C.

The alloy element which is the most effective for improving the low temperature properties is nickel, which, along with manganese, has austenite or face centred cubic lattice forming tendencies. In fact, when the nickel content is such that the structure of the steel is face centred cubic then there is no ductile to brittle transition, although with decrease in temperature from ambient there is a progressive decrease in impact resistance. For 18 per cent chromium–8 per cent nickel stainless steels in the wrought form the steel is notch ductile at the boiling point of liquid nitrogen (−196° C.).

TABLE I. ANALYSES OF STEEL PLATE MATERIALS

Plate Identity	Type of Steel	Plate Thickness	C per cent	Mn per cent	Si per cent	S per cent	P per cent	Ni per cent	Cr per cent	Mo per cent	Cu per cent	Others per cent
A	Grade NDIV C-Mn	$\frac{7}{16}$ in.	0·14	1·30	0·175	0·026	0·018	0·06	0·025	0·005	0·04	—
B	Grade NDIV C-Mn	$\frac{1}{2}$ in.	0·09	1·15	0·12	—	—	0·08	0·06	0·05	0·11	—
C	Grade NDIV C-Mn	$\frac{1}{2}$ in.	0·09	1·15	0·13	—	—	0·07	0·04	0·04	0·10	—
D	Grade NDIV C-Mn	15 mm	0·12	1·44	0·32	0·017	0·026	—	—	—	—	0·009 N_2
E	Grade NDIV C-Mn	8 mm	0·11	1·40	0·28	0·019	0·024	—	—	—	—	0·009 N_2
F	Grade NDIV C-Mn	9 mm	0·14	1·30	0·46	—	—	0·06	0·06	0·05	0·05	—
G	ASTM A.410 Mn-Ni-Cr-Cu	$\frac{1}{2}$ in.	0·11	0·86	0·16	0·023	0·022	0·80	0·75	0·03	0·62	0·0095 N_2
H	Ni-Mo-V-Cr	$\frac{3}{4}$ in.	0·11	1·01	0·15	0·028	0·010	1·58	0·29	0·28	—	0·11 V
J	T.1 Ni-Cr-Mo-Cu-B	0·841 in.	0·14	1·00	0·20	0·032	0·010	0·83	0·56	0·43	0·27	0·036 V 0·0029 B
K	T.1Ni-Cr-Mo-Cu-B	0·841 in.	0·13	0·76	0·23	0·020	0·010	0·92	0·51	0·42	0·28	0·040 V 0·0029 V
L	T.1 Ni-Cr-Mo-Cu-B	0·841 in.	0·12	0·86	0·25	0·028	0·024	0·85	0·57	0·44	0·27	0·038 V 0·0030 B
M	ASTM A.203 3½% Ni	$\frac{3}{8}$ in.	0·13	0·69	0·30	0·009	0·010	3·25	0·08	0·03	—	—

During the recent years, the Research Laboratory of the Society has been assessing the low temperature properties of several qualities of carbon-manganese steels and low alloy steels developed to have transition temperatures below −50° C. Table I contains the details of a selection of plate materials which have been submitted for tests.

Plate A—a normalised open hearth furnace steel to grade ND IV. This particular sample was not specifically manufactured for service at −50° C., but as the steel is a well known notch ductile type, the results obtained are interesting for comparison purposes.

Plates B, C, D, E, F—various normalised open hearth and electric arc furnace carbon-manganese steels to grade ND IV or similar requirement and within the range of 28 to 32 tons/sq. in. tensile strength.

Plate G—normalised open hearth furnace low alloy steel to ASTM A.410.

Plates H, J, K, L—open hearth furnace low alloy steels of a high tensile strength, high yield ratio type, Plate H is a 1½ per cent nickel - molybdenum - vanadium - chromium quenched and tempered steel while plates J, K and L are manufactured to T.1 constructional steel specification and contain boron. All these plates were in the water quenched and tempered condition.

Plate M—electric arc furnace steel to ASTM A.203, containing 3½ per cent nickel, normalised 860° C. and tempered 630° C.

The programme of work carried out in the Research Department consisted of tear tests, plain and notched tensile tests and Charpy V-notched impact tests carried out within the temperature range of ambient to −100° C. The ductile to brittle transition curves obtained from the tear tests and

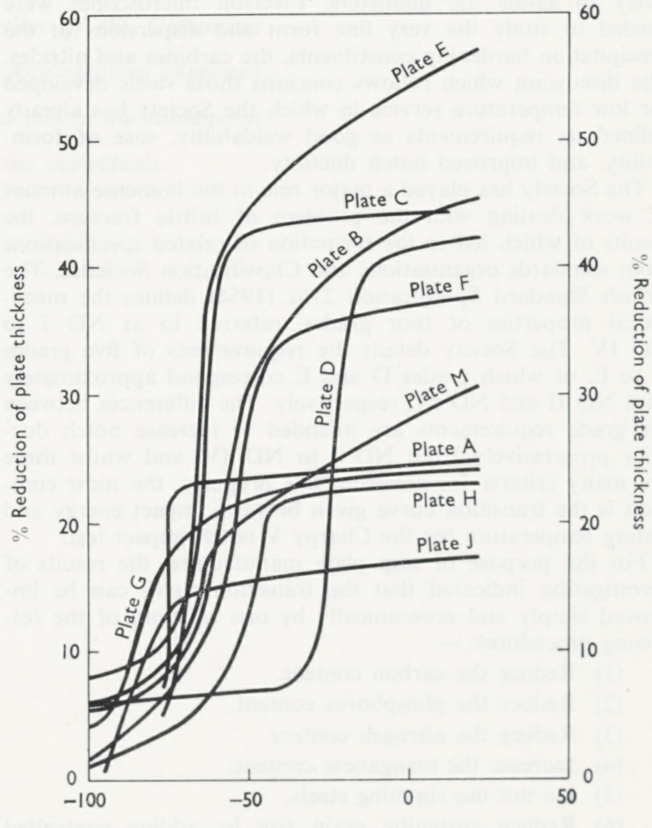

FIG. 1

C-Mn and low alloy steels. Navy tear tests.

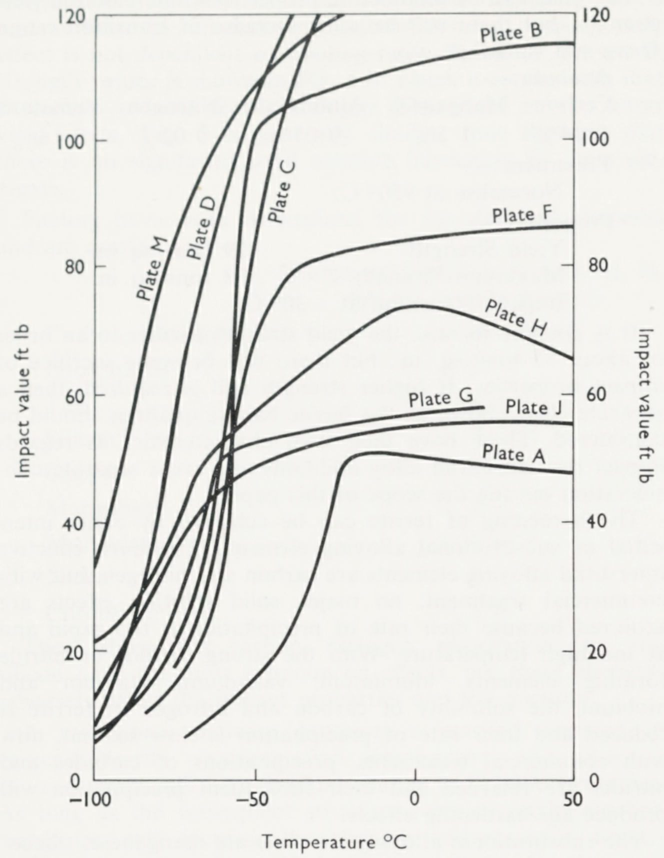

FIG. 2

C-Mn and low alloy steels. Charpy impact tests.

the Charpy impact tests are shown in Figs. 1 and 2. Summarising the data obtained from these tests:—

(1) The static test results on plate A indicate a ductile brittle transition temperature at about −65° C., which is considerably lower than that shown by the impact test results. The latter exhibited a transition at about −20° C. with 15 ft./lb. energy value at −50° C.

(2) For plates B to F inclusive in thicknesses up to ½ in., both static and impact tests indicated transition temperatures between −60° C. and −70° C. The energy values obtained from the tear and impact tests were very high for the ductile condition and transitions indicated by ductility values or the appearance of the fractures were very abrupt. For the one sample greater than ½ in. thickness, the transition was considerably higher at −20° C. to −25° C.

(3) The static and impact tests on plate G revealed a transition temperature at about −75° C. In the ductile condition the impact energy values for plate G were considerably lower than that for the carbon manganese steels.

(4) Static and impact tests on plate H also revealed a transition temperature at about −75° C.

(5) The tear and notched tensile tests on the T.1 steels showed a transition temperature of −80° C. to −85° C.: the impact test results indicated a somewhat higher value between −75° C. to −80° C.

(6) The static test results on plate M—the 3½ per cent nickel steel—indicated a ductile to brittle transition temperature of about −75° C. although the impact test results showed the transition to be about −100° C. The impact energy values obtained on the nickel steel plate in the ductile range were very high.

In the aim to obtain a better combination of desired mechanical properties, particularly with respect to the level of yield strength and impact resistance, there are many laboratories who are carrying out parts of a correlated programme of research to establish the relationships between analysis, heat treatment variables, microstructure and mechanical properties. The controlling factor of the properties is the microstructure and much of the investigation work has been concerned with the marked influence which small additions of alloying elements have on the mechanical properties of commercial steels. The understanding of the metallurgical factors, whilst far from complete, could never have been attempted without the application of newer techniques which are used by metallurgists. These steels consist essentially of a very fine ferrite grain and areas of pearlite, and a part of the study has been the assessing of the proportion of these phases and the grain size. The other metallurgical factors consist of the presence of precipitation hardening constituents of which there are two types, one involving the interstitial elements carbon and nitrogen: the other, the substitutional elements which are in order of their effect, phosphorus, silicon, molybdenum, manganese, copper, nickel and chromium. In these studies, tensile strength itself has not been regarded as important and in any case, can easily be increased by increasing the carbon content. However, this does not have the equivalent effect on the yield point and where design is based upon yield strength, other factors are of more importance. Fig. 3 illustrates the range of impact transition temperatures with respect to the range of yield strength which can be achieved by grain refinement and solid solution or precipitation hardening techniques.

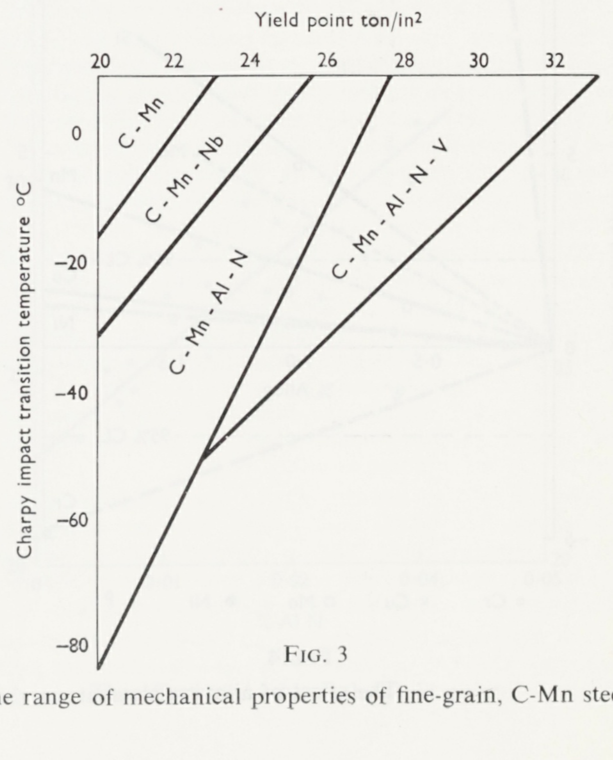

FIG. 3

The range of mechanical properties of fine-grain, C-Mn steels.

Taking the average balanced carbon-manganese steel, its yield strength can be increased by raising the manganese content because the grain size is finer and some solid solution hardening is obtained. Further increases in strength for a balanced composition can be achieved by the addition of niobium or vanadium, and the amount of the order of 0·02 per cent niobium or 0·05 per cent vanadium will achieve adequate properties. Amounts depend upon the combination of the properties required and the steel will require a normalising treatment. A typical steel is:—

Analysis:—

Carbon	Manganese	Silicon	Niobium
0·17%	1·35%	0·07%	0·025%

Treatment:—

Normalise at 950° C.

Properties:—

Yield Strength	24 tons/sq. in.
Maximum Strength	36 tons/sq. in.

Impact Transition at −20° C.

With the steel in the killed condition the scope for minor additions is very broad. Very fine grain sizes are obtained by aluminium or nitrogen additions, viz.:—

Analysis:—

Carbon	Manganese	Aluminium	Nitrogen
0·15%	1·5%	0·1%	0·02%

Treatment:—

Normalise at 950° C.

Properties:—

Yield Strength	25 tons/sq. in.
Maximum Strength	34 tons/sq. in.

Impact Transition at −65° C.

This steel can be modified in properties to increase the yield strength, but there will be some increase of transition range. If we add vanadium, viz.:—

Analysis:—

Carbon	Manganese	Aluminium	Nitrogen	Vanadium
0·14%	1·5%	0·05%	0·02%	0·16%

Treatment:—

Normalise at 950° C.

Properties:—

Yield Strength	29 tons/sq. in.
Maximum Strength	38 tons/sq. in.

Impact Transition at −30° C.

It is possible to raise the yield strength further to an order of about 35 tons/sq. in., but there will be some sacrifice of impact properties. If higher strength still is required, then a separate class known as the ferrite-bainite qualities should be considered. These have their own characteristics as regards impact resistance and alloy additions and are a separate consideration outside the scope of this paper.

The hardening of ferrite can be achieved by either interstitial or substitutional alloying elements. The most effective interstitial alloying elements are carbon and nitrogen, but with commercial treatment, no major solid solution effects are achieved because their rate of precipitation is too rapid and at too high temperature. With the strong carbide or nitride forming elements, aluminium, vanadium, titanium and niobium, the solubility of carbon and nitrogen in ferrite is reduced and their rate of precipitation is slow so that, now with commercial treatments, precipitations of carbides and nitrides are retarded and their subsequent precipitation will produce age-hardening effects.

The substitutional alloying elements are manganese, silicon, phosphorus, molybdenum, nickel, copper and chromium,

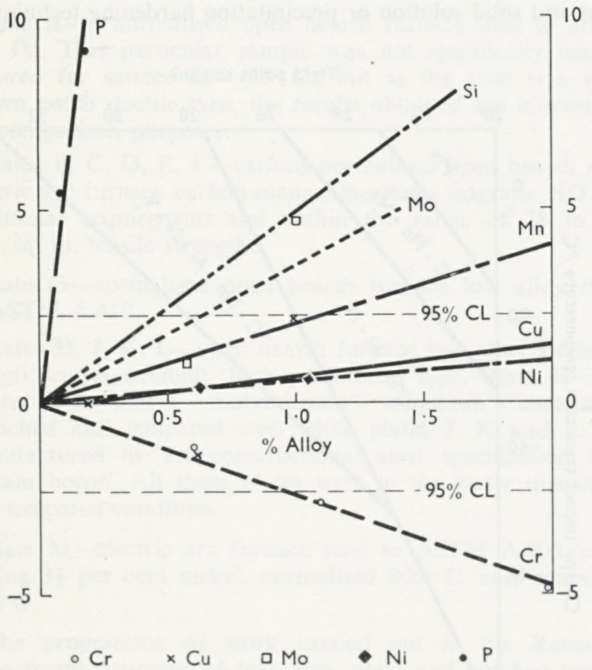

FIG. 4

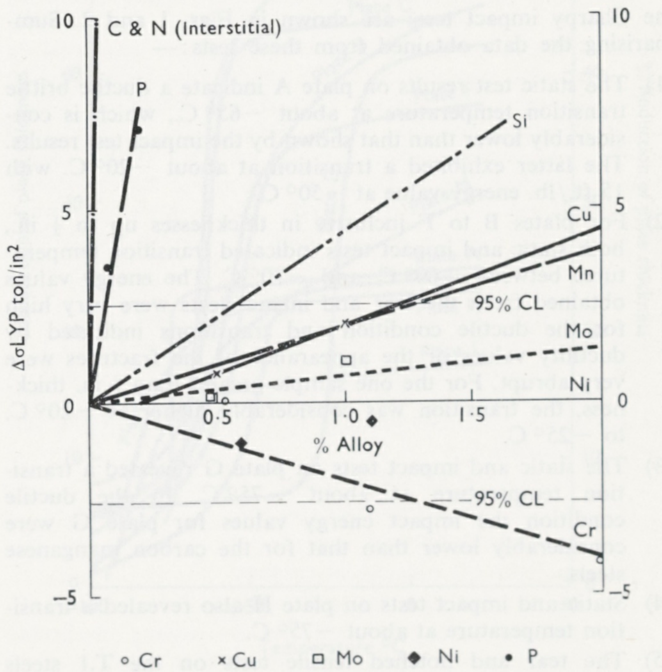

FIG. 5

The effect of alloying elements on the mechanical properties of ferrite-pearlite steels.

which have relatively small solid solution hardening effects and as they do not form precipitates (except copper), their effect is not dependent on cooling rates. The effect on tensile strength values is shown in Fig. 4 in which it can be seen that chromium reduces tensile strength. The effect on the lower yield stress, Fig. 5 is generally similar, thus showing that there is no significant solid solution hardening by these elements.

Factors have been determined for the different elements and are quoted as: —

	Change of Yield Stress tons/sq. in. for 1% weight increase
Carbon, Nitrogen	+280
Phosphorus	+ 55
Tin	+ 8·6
Silicon	+ 5·4
Copper	+ 2·5
Manganese	+ 2·1
Molybdenum	+ 0·9
Nickel	0
Chromium	− 2·0

These show that carbon, nitrogen, phosphorus, tin and silicon have the most pronounced effects, but these elements adversely affect the impact properties and the ductility.

A manganese increase produces a marked decrease in austenite/ferrite transformation temperature, and above 1 per cent there will be a gradual change to bainitic structures. As long as the subsequent structure remains ferrite-pearlite, manganese will increase the strength and maintain impact toughness, in fact, from 18 to 25 tons/sq. in. for the yield strength. With ferrite-bainite, whilst the tensile and yield strengths increase markedly, the impact resistance decreases.

Silicon increases the transformation temperature, coarsens the ferrite grain, and so long as the structure is ferrite-pearlite, the tensile and yield strengths will increase very markedly, although above 32 tons/sq. in. yield strength, ductility will be markedly decreased. With higher manganese and ferrite-bainite structures, silicon increase has very little effect.

Nickel, in increases from 0 to 2 per cent, maintains the ferrite-pearlite structures, increases the tensile strength and the yield strength to a less degree. At 3 per cent nickel, the transformation temperatures are depressed sufficient to give partial bainite in the structure.

Molybdenum having little effect on transformation temperatures, but more a tendency to give split transformation and some bainite, thus whilst increasing the tensile strength, has little effect on the yield strength and a deleterious effect on the impact properties.

If we add vanadium to the steel, the first essential is a high normalising temperature so that full solution of vanadium carbide is obtained. Such will produce coarse grained structures and appreciably decrease the transformation temperatures. But everything depends on the amount of vanadium added. Vanadium is also a grain refiner, and if sufficient is added to counterbalance the coarsening influence of the higher treatment temperatures, then marked increases of both tensile and yield strengths are achieved with high yield ratios. And since the steels maintain the ferrite-pearlite structures, then the impact properties also remain high.

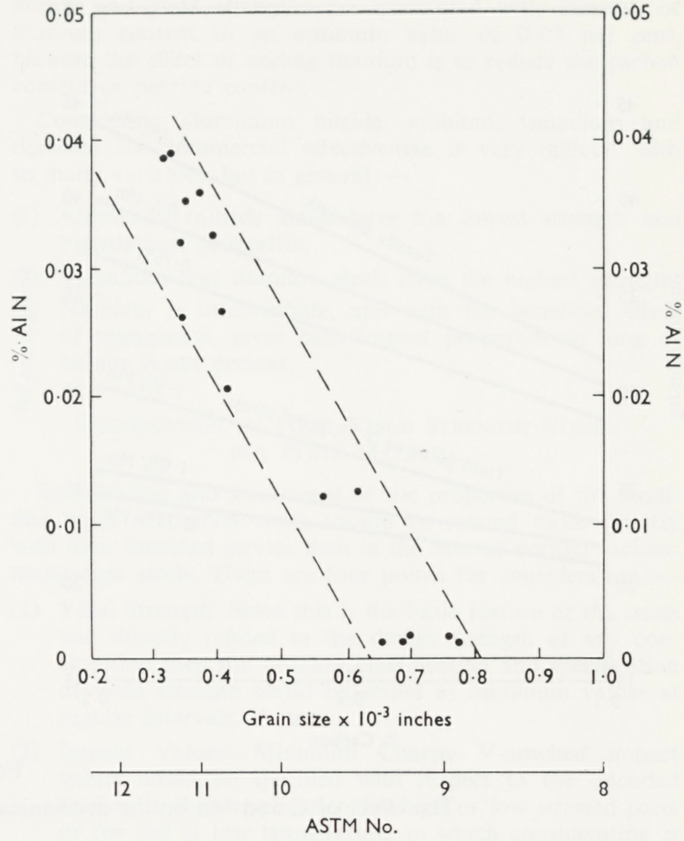

FIG. 6

The relationship between grain size and A1-N content.

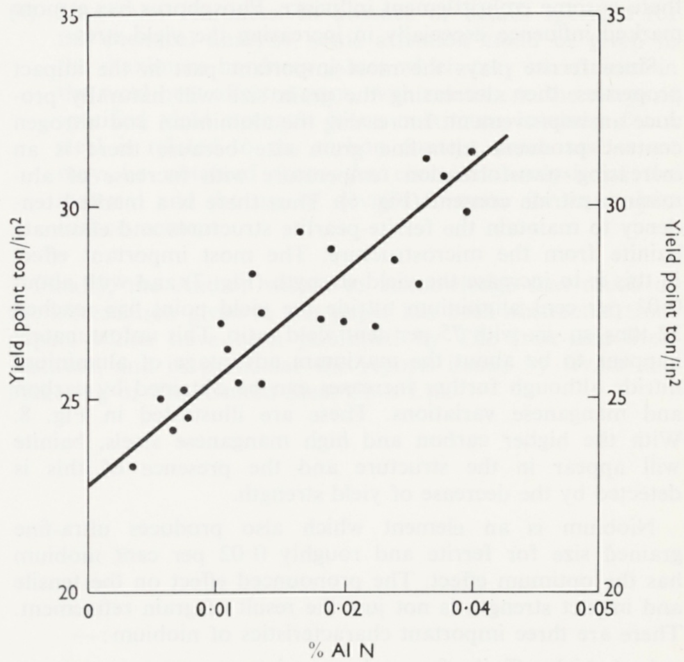

FIG. 7

The effect of A1-N on the yield point.

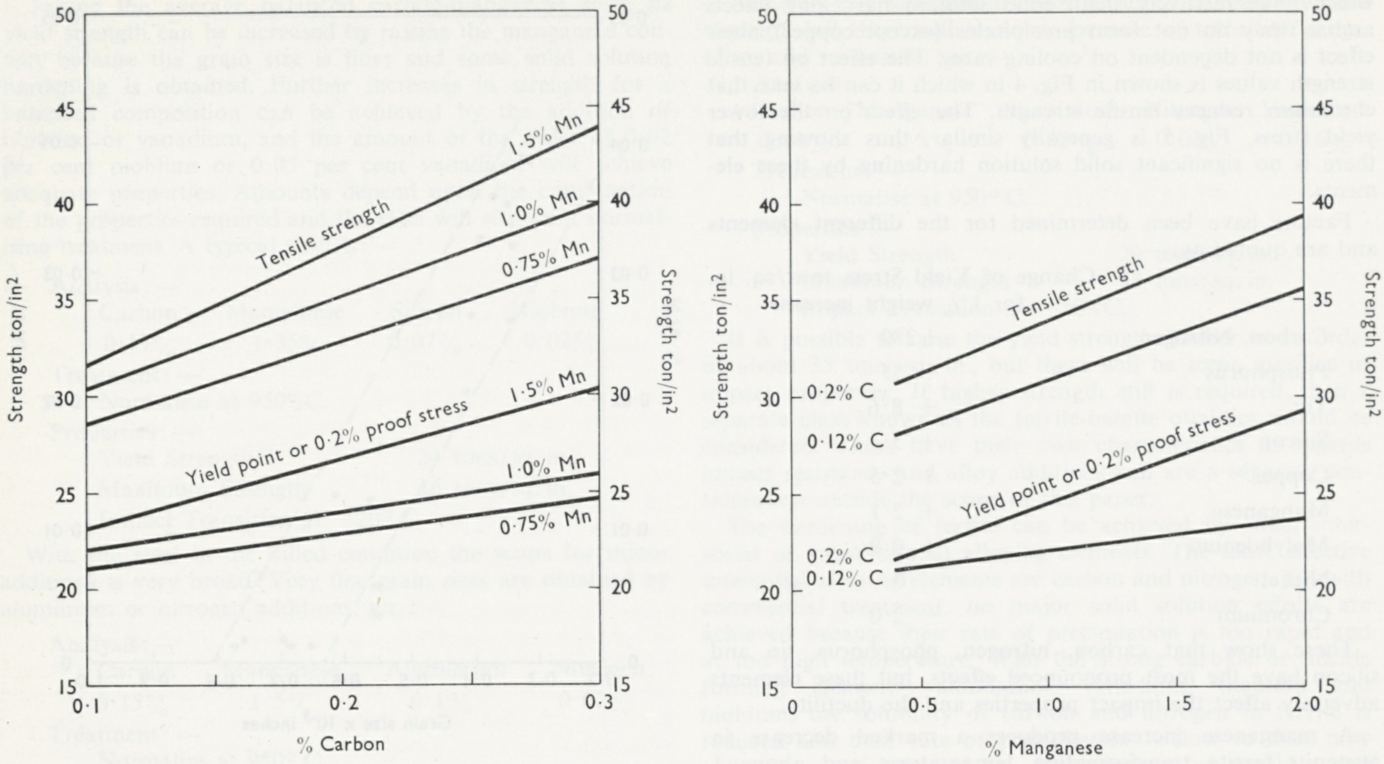

FIG. 8

The effect of C and Mn on the mechanical properties of fine-grain steels (A1-N type).

In carbon-manganese steel, copper has been known to increase tensile strength and corrosion resistance, although there is some embrittlement influence. Phosphorus has a more marked influence especially in increasing the yield stress.

Since ferrite plays the most important part in the impact properties, then decreasing the grain size will naturally produce an improvement. Increasing the aluminium and nitrogen content produces ultra-fine grain size because there is an increasing transformation temperature with increase of aluminium nitride content (Fig. 6). Thus there is a marked tendency to maintain the ferrite-pearlite structures and eliminate bainite from the microstructure. The most important effect in this is to increase the yield strength (Fig. 7) and with about 0·03 per cent aluminium nitride the yield point has reached 32 tons/sq. in. with 75 per cent yield ratio. This unfortunately happens to be about the maximum advantage of aluminium nitride although further increases can be obtained by carbon and manganese variations. These are illustrated in Fig. 8. With the higher carbon and high manganese steels, bainite will appear in the structure and the presence of this is detected by the decrease of yield strength.

Niobium is an element which also produces ultra-fine grained size for ferrite and roughly 0·02 per cent niobium has the optimum effect. The pronounced effect on the tensile and impact strength is not just the result of grain refinement. There are three important characteristics of niobium:—

1. A high affinity for carbon and treatment temperatures exceeding 1200° C. are needed to take the niobium carbide into solid solution in the austenite. On cooling, the niobium

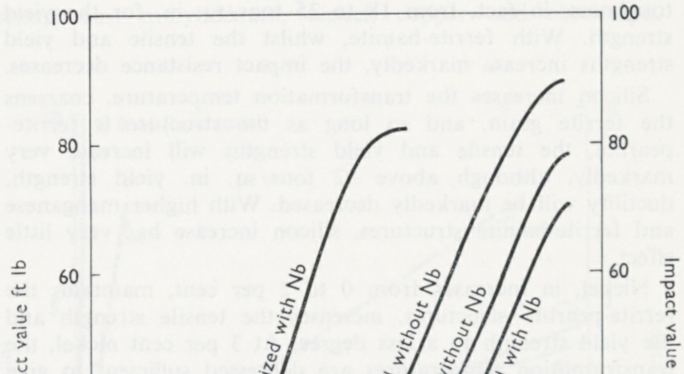

FIG. 9

Effect of 0·01 per cent Nb on V-notch Charpy impact properties of 1 in. thick carbon steel plate in the as rolled and normalised condition.

carbide precipitates over a range of temperature at about 700° C. This precipitate remains coherent with the lattice structure and consequently has a very pronounced strengthening effect. This precipitate is also very stable so that its effect is maintained even after prolonged tempering or stress relief treatments.

When reheating for normalising or quenching treatments, the niobium carbide separates during the ferrite to austenite transformation and tends to restrict any increase in grain size of the austenite. On cooling, the ferrite structure has a very fine grain size. Considering a 1-in. thick plate normalised from 900° C., the effect of 0·02 per cent niobium will be an increase of yield strength of 1–4 tons per square inch, with the ultimate strength less affected and the Charpy V-notched impact transition temperature lowered by 10° C. to 30° C. The following table shows the influence of 0·01 per cent niobium on the tensile properties, whilst Fig. 9 shows the effect on the impact properties.

Analysis 0·20 per cent C. 0·05 per cent Si. 1·52 per cent Mn.

Niobium	*Condition*	*Yield Stress Tons/sq. in.*	*Maximum Stress Tons/sq. in.*
Nil	As rolled	22·2	37·4
	Normalised	23·0	38·0
0·01%	As rolled	28·7	39·8
	Normalised	25·0	36·6

2. A secondary affinity for nitrogen, but since the carbide and nitride phases are isomorphous and the mixed compound contains only a small proportion of nitrogen, little of the latter is removed from solid solution. Niobium thus contrasts with aluminium which removes the nitrogen as aluminium nitride and is the grain controlling agent.

3. A low affinity for oxygen, which is very important, because it means that niobium can be added to semi-killed steels, still produce its grain control and improved mechanical properties in the steel.

Vanadium also has the marked property of grain refining as well as a solution hardening effect and since the element also raises the transformation temperatures, ferrite-pearlite structures are maintained, but above 1½ per cent manganese there will be some bainite produced. 0·2 per cent vanadium is about the optimum content and its effect on the properties is greater than obtained by either aluminium nitride or niobium. This is due to the dual effectiveness of vanadium, and with the use of higher solution temperatures, further increase of tensile strengths are obtained along with the good impact resistance, so long as bainite is not produced.

Titanium also has the effect of increasing the transformation temperatures and refining the grain to a very fine size. The ferrite-pearlite structures are maintained and increases of tensile and yield strengths are associated with increase of titanium content to an optimum value of 0·03 per cent, because the effect of adding titanium is to reduce the carbon content or pearlite content.

Considering aluminium nitride, niobium, vanadium and titanium for commercial effectiveness is very difficult with so many variables, but in general: —

(1) Aluminium nitride steels have the lowest strength and high impact properties.

(2) Vanadium and titanium steels have the highest strength.

(3) Niobium is intermediate, and with the beneficial effect of manganese, gives high impact properties so long as bainite is not present.

Specification of High Yield Strength Steels for Plate Material

Specification and assessment of the properties of the modified types of carbon steels should be related more directly with their intended service than in the case of normal carbon-manganese steels. There are four points for consideration: —

(1) Yield Strength. Since this is the basic feature of the steels and directly related to the design strength of any construction then the primary classification and specification of yield strength could be stated as minimum values at regular intervals of stress.

(2) Impact Values. Minimum Charpy V-notched impact values could be specified with respect to the intended service, in their application for high or low stressed parts or for use at low temperature, in which consideration is given for the lowest temperature expected in service.

(3) Weldability. For all thick plates the evaluation of weldability could be by some form of restraint bead-weld cracking test.

(4) Maximum Hardness. In absence of details specified for the chemical analysis, some attention could be given to the maximum hardness values which may be attained in the weld heat affected zone, with relation to the range of specified yield strength. Since maximum hardness for a steel is related to the carbon content, some assessment may be obtained from the chemical analysis of the material and the use of devised formulæ for equivalent carbon content.

Finally, the Author wishes to acknowledge that much of the information given in this paper has been abstracted from papers which have been published by The Iron and Steel Institute, and in particular the reports issued by Irvine and Pickering of The United Steel Co.'s Ltd.

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Discussion

on

Mr. R. E. Lismer's Paper

DEVELOPMENTS IN CARBON STEELS

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

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Discussion on Mr. R. E. Lismer's Paper

DEVELOPMENTS IN CARBON STEELS

Mr. C. Beason

The Author deserves our thanks for presenting this useful paper, as many of us are now concerned with steel operating at cryogenic temperatures.

There are a few points, however, upon which I would like to comment.

Figs. 1 and 2 show the ductile to brittle transition curves obtained by Charpy impact and Navy tear tests. Firstly, I would like to know how does one assess from these curves the transition point? Is this governed by design, required-minimum temperature, or is there a minimum impact value to be used as design criterion for all structures and pressure vessels, as would appear from the tests. On plate A it is stated that the transition temperature is about $-20°$ C. which would give an impact value of approximately 35 ft. lb. Whilst on plate B the transition temperature is stated to be $-60°$ C. with a corresponding impact value of approximately 30 ft. lb.

Mr. Lismer stated in another of his papers entitled "The Properties of Some Alloys and Weld Metal Deposits at Low Temperatures" published in the February 1962 edition of "Welding and Metal Fabrication", Lloyd's Register of Shipping requires a minimum impact energy of 40 ft. lb. at the minimum service temperature, I don't think I have seen this figure stated in the Rule book, but is this the figure that would govern the ductile brittle transition temperature?

I also believe that this point would be associated with the percentage crystallinity—an impact energy of 35 to 40 ft. lb. having a crystallinity of 50 per cent.

I wonder if Mr. Lismer would comment on this and state if the properties of impact energy and percentage crystallinity are constant for all steels.

There would appear to be a variance between the transition temperatures determined by tear and impact tests. Would it be true to state the Navy tear tests were originally developed in an attempt to use full thickness of the plate with minimum specimen preparation or is it superior in differentiating whether the initial crack is brittle or ductile or whether the propagation of the crack is brittle or ductile, or changes from brittle to ductile, etc. Would the Author state which method he considers the most accurate for assessing a steel?

On page 3 the Author states that the tensile strength has not been regarded as important as it can easily be increased by increasing the carbon content, which does not have the equivalent effect on the yield point, and where design is based on yield strength other factors are important, would Mr. Lismer enlarge on these factors.

I am not sure how to interpret Fig. 3; if the yield strength is to be used as the basis of the design, is this to be governed by the Charpy impact transition temperature. For example, if a design temperature of $-25°$ C. is to be catered for by the carbon-manganese-niobium steel having a yield point of 26 tons/sq. in. at normal temperature, should the design stress be based on a yield strength of 22 tons/sq. in.?

I have concluded that the transition temperatures are based on an impact energy of 35 to 40 ft. lb.

Thanking the Author once again for this very informative paper.

Mr. Boyd

The paper will be rather over the heads of most of us, including myself, but gives a good insight into the problems facing metallurgists concerned with developing steels to keep pace with progress in engineering.

A word of warning is necessary in relation to the discussion on the "ND" series of steels to B.S.2762. These specifications cannot be regarded as "equivalent" to the unified Grades A, B, C, D and E. Moreover, the figures given in the paper for the analyses of the ND steels should be regarded merely as descriptive, and not as specification values. The actual values for a given piece of steel may differ considerably from those given.

The actual specification values given in B.S.2762 : 1956 are: —

	ND. I	ND. II	ND. III	ND. IV
Carbon, max. %	0·20	0·20	0·17	0·17
Silicon %	—	—	0·10 to 0·35	0·10 to 0·35
Manganese, max. %	1·50	1·50	1·50	1·50
Sulphur, max. %	0·06	0·06	0·05	0·05
Phosphorus, max. %	0·05	0·05	0·05	0·05
Charpy ft. lb. at 0° C.	20	—	—	—
−10° C.	—	—	—	45
−15° C.	—	20	—	—
−20° C.	—	—	—	40
−30° C.	—	—	20	34
−40° C.	—	—	—	25
−50° C.	—	—	—	20

Note: The above Charpy values are minimum averages for three test pieces except for ND. IV, where the values are minimum for individual test pieces. The specification should be consulted for further details.

Similar remarks apply to the transition temperatures cited, which can vary very widely, even in steels to the same specification.

The paper does not define "transition temperature" and this should be clarified. The concept of transition temperature is quite definite, i.e. the temperature (range of temperature) at which the character of the fracture changes from ductile to brittle, or vice versa, but this temperature varies according to the type of test and the constitution of the material. Sometimes, the transition temperature is defined as the temperature at which a certain energy value may be expected in a particular test, such as the Charpy V-notch test. Presumably a definition of this kind has been used in the paper, but the energy value is not stated.

Probably the most important developments in carbon steels in recent years have been: —

1. The introduction of tonnage oxygen in steelmaking.
2. The advances in techniques for taking full advantage of the peculiar properties of certain alloying elements, such as niobium, in small quantities.
3. The development of cheaper steels having higher tensile strengths and yield points.

4. The development of steels which retain their notch ductility down to low temperatures.

It would be very useful if the Author could elaborate the significance of these developments from the engineering viewpoint, rather than the problems they pose for the metallurgist.

The paper is a valuable addition to the transactions, and shows the Author's remarkable grasp of a very complex subject.

Mr. R. D. FRENCH

The paper is an extremely useful one to Engineers as it does give some sort of clue to what a carbon (low alloy) steel is like, given the chemical analysis.

This is all the more acceptable as metallurgists are not usually very forthcoming as regards their secrets. However, since they are a form of wizard (Sheffield water tradition and throwing sticks on forgings proves this) they are fair game for questions and I should like to ask a few minor ones.

(a) How much of the work described was done at Crawley? Does it include all tests shown in Figs. 1–9 inclusive.

(b) Is the test shown in Fig. 1 a true Navy Test, as that normally includes only maximum load, energy requirement start fracture and energy to propagate fracture.

(c) What was the reason for notched tensile tests and what results obtained?

(d) What size and type of test piece and rate of testing was employed for yield point and U.T.S.? Why the lower and not the higher yield point?

(e) Have any corrosion cracking or strain-ageing micro-cracking effects been obtained with the alloys mentioned or are they free from such troubles.

(f) Have tests been carried out on weld metal since the strength of plate is that of its weakest part.

In this connection it should be mentioned that 12–15 per cent of the Charpies in a recent reactor, with similar material to Plate A but approximately 4 in. thick, were under the plate specification 20 ft. lb. at $-10°$ C. for weld metal tests. This also applied to electroslag welds in 14 in. thick ducting nozzles.

(g) As regards the Charpy Tests the authorities do not appear to agree and in 1960 an American, Wenner Goldsmith, stated that the notched bar impact test does not determine fundamental material properties for which other tests are preferable. Also that there is no significance in comparisons between transition curves of low alloy steels of varying composition. Such tests are mainly useful as a check on heat treatment in a known material. Has the Author any further comment on this subject?

Finally there are a few questions at random, but I should like to thank the Author for a very useful addition to the list of Engineer's handy references.

Mr. G. P. SMEDLEY, B.Eng., B.Met.

I am sorry that I was out of London when Mr. Lismer presented his paper. The subject is of special interest at the present time.

Attention has been given to the selection and weldability of low alloy structural steel for well over 30 years. Much of the earlier work is still vital to current development of higher strength steel. Table I shows the coefficients of carbon equivalent proposed by Dearden and O'Neill[1] and based on extensive tests. Columns 1 and 2 contain the divisors which should be used to estimate the carbon equivalent for the calculation of the tensile stress and yield stress of a steel. These are given by the following:—

Tensile stress = 16 + 40 (carbon equivalent %) ± 3 tons/sq. in.
Yield stress = 10 + 20 (carbon equivalent %) ± 2 tons/sq. in.

There is a marked difference between the values given by these formulæ and the data presented by Mr. Lismer on page 5 and in Figs. 4 and 5 of his paper. Obviously there is an error in his assessment of the influence of the carbon content of a steel on the yield stress. Would he review his data and indicate the level of reliability for approximate assessment of the properties of steel?

From the point of view of fabrication of a steel by welding, the level of hardness in the heat affected zone is vital. Hard zone cracking is likely to occur if the hardness is much in excess of 350 VPN. As much of the welding in a large structure involves conditions of restraint, the associated stress is liable to cause major extension of hard zone cracks. The hardness in the zone adjacent to weld metal depends on the composition of the steel and the rate of cooling following deposition of the weld metal. The former is expressed mainly by the "carbon equivalent" of the steel. The factors for the various elements are shown in column 3, of Table I. From the results of Reeve tests on half-inch plate with a 0·045 sq. in. bead size (i.e. a fillet of about a quarter of an inch leg size) the hardness (VPN) of the heat affected zone for steel with a carbon equivalent of from 0·3 to 0·7 per cent is given approximately by:—

Weld hardness = 1200 (carbon equivalent %) − 190.

Thus for normal welding conditions without preheating and for ordinary electrodes, the maximum carbon equivalent of the steel that can be tolerated (for a heat affected zone hardness of 350 VPN) is 0·45; the actual percentage of carbon being 0·2 per cent max. for low alloy steel. However, latitude is necessary to allow for actual site conditions, geometry, thickness, and variation of composition of the steel from ladle analysis. Preference is, therefore, given to a nominal carbon equivalent of the steel of about 0·4 per cent.

During the cooling of a weld the tendency to hard zone cracking and the hardness of this zone is governed largely by the rate of cooling when the temperature is about 300° C. This rate of cooling can be reduced by preheating or by increasing the heat input during welding (i.e. using a larger gauge of electrode). It can also be reduced by lagging or by a change of joint geometry, to one with a smaller heat flow. For a given weld bead size, the rate of cooling obviously increases with increase of plate thickness, hence the recommendations which appear in some Codes for preheating when plate thickness exceeds a particular level depending on the steel.

The tendency to hard zone cracking increases with the hydrogen content of the steel or the weld. In general, therefore, higher hardness can be tolerated without cracking when low hydrogen electrodes are used. On the other hand it must be appreciated that the yield stress and tensile strength of the hard zone increase with the hardness and at the same time the ductility is reduced. This can be corrected by post welding heat treatment (i.e. tempering or stress relieving). With "as welded" structures particularly where the welding involves restraint, the effects of which remain, it is undesirable to have high hardness in the heat affected zone. Any small crevice or service defect could become a major fracture even

[1] J. Dearden and H. O'Neill; Trans. I. of Welding; V3; 1940.

if the parent metal was originally notch tough. The writer has preferred heat affected zone hardness to be less than 350 VPN except for random spots, and where possible less than 320 VPN. The advantages of low hydrogen electrodes should be exploited, therefore, with caution where higher hardnesses may be expected in the hard zone and post welding heat treatment is impractical.

Turning now to the steel, Dr. Reeve has pointed out the aim for structural purposes that the overall cost using a high strength steel should be no greater than for mild steel construction. In consequence cheap low alloy structural steels are essential if they are to be competitive for normal applications. The alloy addition(s) must involve low cost and this eliminates a number of valuable elements which would be acceptable otherwise due to their minor influence on weldability. Attention is therefore focused on additions which Dr. Noren has termed micro elements. These are elements which when present in small amounts in steel increase appreciably the yield stress and the tensile stress. Acceptable elements should not adversely affect formability, weldability and notch toughness.

Mr. Lismer has considered the influence of a number of micro elements on the strength and impact properties of steel. It is quite clear that the steel must be carefully controlled during hot working and by heat treatment to develop the necessary properties and to avoid detrimental effects. For these reasons I understand and perhaps Mr. Lismer would confirm, that the main of these additions are generally unsuitable for cast steel as the constituents tend to migrate to the grain boundaries and cause embrittlement.

With reference to weldability, Swinden & Reeve[2] showed in 1938 that titanium could be slightly beneficial. It combined with some of the carbon in the steel, thereby reducing the hardness in the heat affected zone for given basic steel composition and welding conditions. More recent work indicates that niobium has a similar effect. However, I have heard of some site welding difficulties with a higher strength steel containing vanadium.

While the benefit of titanium and niobium can be appreciated, I have no information on the role of nitrogen in the welding of the high nitrogen steels. Has Mr. Lismer any data on this subject? I should also like to have his views on the influence of some of the micro elements on the properties of the steel in the heat affected zone for a hardness within the range of 300 to 400 VPN. An element such as vanadium can render a steel sluggish to respond to heat treatment and make it more prone to failure under restraint if a defect is present or develops during service. Are these features likely to be of consequence for any of the alloys to which he refers in his paper?

Mr. J. Naysmith

The Author is to be congratulated on this excellent and valuable paper which will contribute much to the understanding of notch ductile steels.

Carbon and Manganese

The optimum combination of tensile and impact properties of normalised steels. at both normal and low temperatures, can be obtained by using the lowest carbon content compatible with obtaining the tensile strength required. In addition

[2] T. Swinden and L. Reeve; Trans. I. of Welding; 1938.

TABLE I

TENSILE, YIELD AND WELDABILITY COEFFICIENTS OF CARBON EQUIVALENT (APPROX.)

Element	Range	Carbon Equivalent		
		Tensile	Yield	Welding
C	0·1–0·7%	1	1	1
Si	up to 1·5%	6	3·5	—
Mn	0·4–0·8% with 0·2% C	8		
	0·4–0·8% with 0·3% C	7		6
	1·3–1·8% with 0·1–0·28% C and ·0–0·5% Cu	5·5	4	
Cr+Mn	0·3–0·5% Cr; 0·8–1·2% Mn	6	4·5	5 (Cr) 5·5 (Mn+Cr)
P	below 0·06%	0·8	0·8	21
Ni	1·8–4·8% using Mn/8	16	—	15
	1·8–4·8% using Mn/6	—	9	
Cu	below 0·5%	∞	∞	∞ (0·5–1·0%; 13)
Mo	0·4–0·65% using Mn/8	10	—	
	0·4–0·65% using Mn/6	—	3·5	4
V	0·16–0·19%	0·6	0·35	5
Co	2·3%	16	10	150

to raising the impact transition temperature ranges of pearlitic structures, increasing carbon content lowers the maximum impact energy values and widens the temperature range of the transition.

The effect of carbon content on the impact properties of fully hardened and tempered medium carbon steels depends to some extent on the hardness obtained after tempering. It is generally accepted that at very high tensile levels the lowest possible carbon level giving the necessary strength is desirable.

Changes in transition temperature of over 50° C. can be produced by changes in the chemical composition or microstructure of mild steel. The largest changes in transition temperature result from changes in the amount of carbon and manganese (Rinebolt and Harris). The 15 ft. lb. transition temperature for Charpy V-notch specimens is raised about 15° C. for each increase of 0·1 per cent carbon. This transition temperature is lowered about 5° C. for each increase of 0·1 per cent manganese. Increasing the carbon content also has a pronounced effect on the maximum energy and the shape of the energy transition curves. The Mn:C ratio should be at least 3:1 for satisfactory notch toughness. A maximum decrease of about 50° C. in transition temperature appears possible by going to higher Mn:C ratios. The practical limitations to extending this beyond 7:1 are that manganese contents above about 1·4 per cent lead to trouble with retained austenite while about 0·2 per cent carbon is needed to maintain the required tensile properties.

Increase in the manganese content refines the grain and fine grained steels are less brittle than coarse-grained steels. Manganese is a mildly carbide forming element. Increase in the manganese content changes the composition of carbides and alters their behaviour. Manganese favours the spheroidization of carbides which for high manganese content form globules instead of bands.

Adding manganese to iron alloys of low carbon content decreases the amount of cementite film surrounding pearlite grains. The thickness of the carbide film is diminished and the pearlite structure is refined. Manganese present in the ferrite has a higher affinity for carbon and nitrogen than has iron and therefore manganese increases the solubility of carbon in ferrite, it also affects the response of steel to heat treatment, and also diminishes the ageing tendencies of steel.

Silicon

The effect of Silicon on the low temperature impact properties is only slight. In the role of a deoxidiser it is beneficial but as an alloying element it is detrimental, in a low carbon semi-killed steel containing 0·8 per cent manganese. Banta, Frazier and Lorig reported that the Charpy impact transition temperature was lowered by increasing the silicon content up to 0·16 per cent, but that with further additions the transition temperature increased.

Nickel

Nickel is generally accepted to be beneficial to notch toughness in amounts up to 2 per cent and seems to be particularly effective in lowering the transition temperature. The most generally useful nickel bearing steels are those in which carbon and manganese are kept low to minimise the retention of austenite and to avoid undue hardenability. In a typical example of this type of steel containing 0·02 per cent carbon the addition of 3·6 per cent nickel lowers the transition temperature by 33° C., the lower transition is reported to be lowered 3° C. and the upper lowered about 0·5° C. per 0·1 per cent nickel added.

AUTHOR'S REPLY

I should like to thank my colleagues who have taken part in the discussion and added so admirably to the value of my paper by their own considerations of the subject.

Mr. Beason and Mr. Boyd have raised the concept of "transition" in steel, the former defining a change from ductile to brittle characteristics, the latter defining the "temperature level" at which this change occurs. "Transition" depends on the nature of the test, and the Society has in its requirements demanded minimum energy values given by the Charpy V-notch impact test at a certain sub-zero temperature. This in no way indicates a "transition" and some steels with above say, 50 ft. lb. impact energy can demonstrate a crystalline fracture. This is because the energy value is the sum of energy to initiate a crack from a certain notch contour, and the energy to propagate that crack. Experience has taught us that for ship plate materials, 35 ft. lb. minimum impact energy at the minimum service temperature can be the difference level between ductile and brittle failure in service. Other specifications demand other minimum energy values as indicated in the table contained in Mr. Boyd's contribution.

"Transition" to the metallurgist is a change in the mode of fracture of the ferrite grains which form the bulk constituent of plate steel. Pearlite, the second constituent, has only a small part to play, mainly in the role of initiation of cracks at the ferrite-pearlite interfaces. Ferrite (or pure iron) crystals have a body-centred cubic structure and at room temperature are soft and plastic. When overstressed, they deform by a slip mechanism, the direction of the slip being the cube diagonal, and the plane of slip may be one of a number of planes whichever happens to lie in the direction of the applied stress, or on which the resolved shear stress is the greatest. The crystals now begin to deform and fracture occurs when a certain critical shear stress in the slip direction is exceeded, because during deformation the slip planes begin to take on a definite orientation.

At lower temperatures, the ferrite crystals will only behave plastically when two of their crystal planes, referred to as [111] and [110], are orientated in the direction of the stress. If the [100] plane approximates to the direction of stress, the crystal will cleave sharply and fracture without prior deformation, i.e. entirely brittle. The word "approximates" includes an inclination of up to 20° from the direction of the stress.

Obviously, where there are many crystals, some will have planes which will be more favourably orientated to the direction of stress, and at the lower temperature cleave whilst the others will re-orientate. The fracture will thus propagate in stages, choosing the least line of resistance which will be the [100] planes suitably orientated, but at the same time initiating cleavage fractures in neighbouring crystals which otherwise would have deformed. The reason for the latter lies in the stress conditions which exist at the end of a crack in a stressed metal. When such stress concentration exists, the strain in the direction of the crack produces transverse tensions and gives rise to a state of triaxial tension which makes it more probable

that cleavage will precede slip on all planes, irrespective of their orientation.

There are complexities which influence this mode of fracture, and these would need many pages to set out. From the engineers' point of view, "transition" is the concern of a piece of steel with a certain depth and sharpness of notch, which at some temperature level fails in a brittle manner when shock loaded. Above this level, the shock will cause local deformation. Below, the shock can initiate a crack which requires only a fraction of the yield stress of the material to propagate. The aim of the laboratory testing procedure is to mimic such behaviour, for which reason the testing of the full section of the plate is more logical. The Charpy impact test is a very severe test which can classify structural steels according to their metallurgical variations, i.e. analysis, manufacture and treatment. The only precaution for the engineer is that the minimum service temperature is above that temperature at which the impact energy value is obviously low and the fracture very predominantly crystalline. For the account of the considerations by the Society towards minimum acceptable energy values, Mr. Beason is referred to the papers by Mr. Boyd, and I can only excuse my reference to quoting the minimum energy value at 40 ft. lb. in February, 1962, in that at the time of preparing my previous paper, I was advised by the "powers", that this was their consideration.

Mr. Beason should not be too critical in studying Charpy impact energy—temperature curves, which generally demonstrate broader transition temperature ranges when compared with graphs plotted from data obtained from other types of low temperature tests, in which the criterion for examination is fracture appearance. These other forms of testing would be preferable if only they could be universally adopted for steel plate production testing. Whether the tear test (as one alternative to the Charpy test) is the most "accurate for assessing a steel" is difficult to comment on because unless such tests are made on plate materials which had "failed" in service under known conditions, and in which the service fracture were recorded, there is no basis for comparison. However, tear test procedure can be very useful and especially in the study of weld deposits.

The enlarging for Mr. Beason of the considerations by which the metallurgist concentrates on yield stress as the important factor, and the elaboration of the points of the carbon steel development given in Mr. Boyd's contribution create a mammoth task.

The reasons for the present day impetus on carbon steel development are:—

1. Economical considerations for both the steelmaker and the shipowner.
2. New outlook on the micro-structures of steels and appearances of fractures produced by new techniques of examination, e.g. electron-microscopy, transformation studies, etc.
3. New concepts in fracture mechanics concerned mainly with the nature of the initiation and propagation of brittle fracture.
4. New theories dealing with yield point phenomenon and strain hardening mechanisms in iron and steel.
5. Significant aspects of dislocation theories in the behaviour of ferrite crystals under stress.
6. Very detailed analysis (mainly statistical) of the metallurgical parameters which influence the mechanical behaviour of steels in comparison with the properties of pure iron.

Research has concentrated on yield strength because the examination of service failures in large welded structures has shown little evidence of plastic strain. It is therefore necessary to be able to predict behaviour of the steels in the non-ductile condition, and experiments have shown that the brittle strength is related to the yield strength. Where some variety of notch effect is present, brittle strength is greater than yield strength, and since for brittle fractures no plastic strain at the root of the notch is evident, the existence of strain hardening accounts for the higher stress needed to initiate a cleavage fracture.

With regard to what was said in my paper on the relation of carbon content to tensile and yield strengths, this can be further explained. Dealing with ferrite-pearlite steels only, increasing the carbon content will increase the pearlite constituent and thus proportionally increase the tensile strength. Initial plastic deformation in these steels is confined to the ferrite grains, and thus the yield stress is not affected by the amount of pearlite. This does not mean that carbon has no effect on the yield point. It has, but in a different context to its influence on the tensile strength. Carbon will act as a grain-refiner or provide the means for the formation of carbide precipitates and these two features affect the yield strength.

I am sorry if Fig. 3 of the paper is a little misleading, but there is no attempt in the diagram to indicate yield strength at different temperatures. The yield strengths are those given by normal testing procedures. If we have a design stress which requires a steel of yield strength of 21 tons/sq. in., then impact transition for C-Mn steel will be about −10° C., for C-Mn-Nb steel about −25° C. and for C-Mn-Al-N steel about −70° C. For a yield strength of 24 tons/sq. in., C-Mn steel would have a transition temperature above ambient, C-Mn-Nb steel about −5° C., C-Mn-Al-N steel about −40° C. and C-Mn-Al-N-V about −45° C. Requirements of over 26 tons/sq. in. yield strength need the extra alloys with range of impact transition temperature as indicated.

Further research work has enlarged the scope of these properties, and at present, there are steels containing small amounts of niobium, vanadium and titanium, but in optimum quantity, which have a range of yield strength and transition temperature from 21 tons/sq. in. with −100° C. to 36 tons/sq. in. with −60° C.

The introduction of tonnage oxygen in steelmaking has no significance with respect to the brittle fracture problem and is only a necessity in the pattern of demand for cheap steel and higher output rates. After the last war, the changes from coal as the base for fuel to alternative fuels such as crude oil or pitch, could not be applied to the Bessemer converter method of steelmaking. Further, on the Continent, the scarcity of lower-phosphorus ore in the Lorraine area and the necessity to expand the steel industry with the minimum of available capital, enforced an economic situation which was overcome by using oxygen as the fuel to give steel of satisfactory quality. Meanwhile, in the U.K., developments were based on the open-hearth furnace and oxygen-blowing was applied to enrich the contents of the flame. There was some prejudice against basic Bessemer steel. Later, however, the U.K. had raw material problems. Steel scrap is the bulk charge of the open-hearth furnace, and available supplies were insufficient for expanding output. The development of massive iron-ore fields in the more-or-less undeveloped parts of the world and the progress of in-bulk transportation and handling facilities, now meant that the

oxygen-blown Bessemer process offered the higher rate of production for good quality steel.

I am grateful to Mr. Smedley for enlarging on the points of carbon equivalent factors for incidental or alloy additions to carbon steel. In view of present-day discussions within the Society, I had been careful to avoid commenting on the significance of the many formulæ for estimating steels for constructional or engineering purposes. The Dearden and O'Neill formula is the most well-known in the U.K. and basically the formula and application in welding is as follows:—

$$C.E. = C + \frac{Mn}{6} + \frac{Ni}{15} + \frac{Cr}{5} + \frac{Mo}{4} + \frac{Cu}{30}$$

With C.E. $<$ 0·4—most normal methods of welding can be carried out

C.E. of 0·4 to 0·6—Pre-heat before welding

C.E. $>$ 0·6—Pre-heat and post-heat, treat with any other practical precautions.

The Dearden and O'Neill formula will thus judge some criterion of weldability of the steel because it is related to maximum hardness values obtained under quenched conditions, or the production of martensitic types of micro-structure. Such structures occur in the heat-affected zones of weldments. The data given in my paper are only in association with ferrite-pearlite structures produced by the normalising treatment of the steel.

The Dearden and O'Neill formula can, however, only give the "probable" maximum hardness, because size of weld, plate thickness and pre-heat must also be taken into account. The extent to which hard-zone cracking can occur in steels where carbon equivalent or maximum hardness is calculated to be above "desired" values can be alleviated as indicated by Mr. Smedley: pre-heating, delayed cooling, use of low hydrogen or austenitic electrodes, etc. All these are associated with the permeability and amount of hydrogen in the weld zone, the temperature at which the austenite/martensite transformation takes place and the extremely low ductility of the martensitic structures.

The full significance of the thermal and stress cycle across a weld zone is highly complex, and the addition of alloys to a steel will increase the tendency to crack because they decrease temperature of the austenite/martensite transformation. They may, however, produce a finer grain structure which would tend to reduce the cracking tendency. In general, if the carbon content is reduced, the weldability of the steel is improved, but stated more metallurgically, if the solubility of carbon in austenite is lower, then from the phase-diagram aspect, the extent of the austenite phase field is decreased and the temperature at which austenite transforms is raised. Small additions of carbide-forming elements such as niobium vanadium, titanium and zirconium have such an effect on austenite and their carbides remain out of solution in the austenite at normal normalising temperatures. For reasons set out in my paper, niobium is the most effective of these elements, but much also depends on the hot mechanical working procedure for the steel, particularly roll-finishing temperature. For this reason, niobium additions to cast steel do not have the same optimum or beneficial effects as in rolled steel. Mr. Smedley also mentions the difficulties of welding high strength steel based on vanadium additions. Without entering the realm of metal physics it is difficult to explain the finer points of difference between niobium and vanadium. Calculations of the frictional stress in these steels have shown that vanadium is only equally effective as niobium in increasing the yield stress if the nitrogen content of the steel is considerably higher. This is because, whilst niobium is present as a carbide, vanadium precipitates as the nitride. The latter is thus less effective with relation to the carbon content of the austenite and weldability, and substantial grain coarsening occurs at lower heat treatment temperatures used for the vanadium steels. On the other hand, vanadium is more easily controlled in steel compared with niobium.

With the extra care needed and given in fabrication and inspection of weldments using the high tensile steels, I would expect much fewer "faults" to develop under assessed service conditions compared with equivalent structures of normal carbon-manganese steels. The low transition temperature in all section size with the extra safety factors given by high yield strength are very desirable features against the probability of stress raisers from welding.

Mr. French is thanked for his remarks and minor questions, which unfortunately have no minor answers. The work reported in my paper has partially been the result of tests carried out in the Society's Research Laboratory and partially the work of the British Iron and Steel Research Association and other Research Departments of the steel plate producing firms with which the Author is privileged to have close contact. The Navy tear test which originated in U.S. was first copied and modified by the Society's Research Department in 1956/57 to study the behaviour of the aluminium alloys intended for the tank construction in the *Methane Pioneer*. For the significance and results of notched tensile tests on the series of steel, I would refer Mr. French to the papers issued by "Welding and Metal Fabrication" in January and February, 1962. In the tensile testing on these materials, the specimen size is $\frac{1}{8}$ sq. in. section area and 2 in. gauge length. The rate of straining is standardised at 0·054 in. per minute. A stress-strain curve is obtained and from the nature of the curve at the lower yield point, the work hardening factors for the steel are assessed using well-known formulæ. A great amount of work, including low temperature testing, has been made on welded plate material. Data is available within the Society's records for Mr. French to examine and specific queries can be discussed. Data is also available on work carried out by other research bodies on the aspects of corrosion and strain-ageing problems. Much literature has also been issued on the significance of the Charpy impact test for assessing plate materials and relevant weld deposits. Notched bar impact energy values do not give fundamental properties in terms which are a necessity for design engineering, but knowing the worst service conditions likely to be imposed on any structure, there are broad significances between steels with Charpy impact transition temperatures below or above those expected in service.

I am very grateful to Mr. Naysmith for the further data in support of my paper and look forward to a paper from himself giving the more detailed assessment from his own studies of the effects of the various elements on the tensile and impact properties of normalised structural steels.

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STEERING GEAR

by

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STEERING GEAR

By R. G. Lockhart

It is now 27 years since Mr. G. Buchanan read a paper[1] on this subject. The paper was in three parts, the first two parts on steam gear and the third on electro and steam hydraulic gears. The findings of the Steering Gear Committee's report on rod and chain gears then recently published were dealt with and although electro hydraulic gear was a very recent innovation, the Author expressed enlightened knowledge of this part of the subject. If age has outdated this paper, Mr. Buchanan has not lost his interest in the subject, as it was principally at his suggestion that this effort was conceived.

In presenting these thoughts it is hoped they might form a basis for bringing the requirements of the Rules a little more into line with current practice and to this end the comments and criticism of colleagues are invited.

HISTORICAL

Until the 13th century B.C. there is no evidence to suggest that steering was effected other than by hand paddles. About this time fixed paddles controlled by handles or helms were in use in various positions and eventually settled down to the starboard or steering side. The ability of side paddles to manœuvre could not be questioned by anyone who has seen a paddle ship guided to the pier. This type of paddle steering was in general use until about the 12th or 13th century A.D. when the stern rudder controlled by a hand tiller was introduced. As ships grew the hand tiller was more difficult to operate and block and tackle gear was the next step in the evolution. The hand winch may have appeared almost simultaneously and with block and tackle was probably the fashion until the end of the 17th century.

The 18th century saw the introduction of the steering wheel and drum but soon the quadrant replaced the drum. The Napier differential screw gear was also developed in this era. Right and left hand screw gear carrying travelling nuts which were linked to the tiller were the basis of this gear. As the screw was turned the nuts moved to and from each other thus moving the tiller. This is the type of gear referred to in paragraph D 2702 of the Rules and is shown in Fig. 1.

It is recorded that in the 19th century the steering of a man-of-war was effected by about 100 men applying themselves to the hand tillers of the capstan or winch.

The first notable introduction of steam was in the *Great Eastern*[2], the added wonder of this gear was the incorporation of a follow-up system or hunting mechanism. This is illustrated in Fig. 2 and it is thought worth offering a brief description of the mechanism. As control shaft C is turned

1 *et seq.*—For references see page 16.

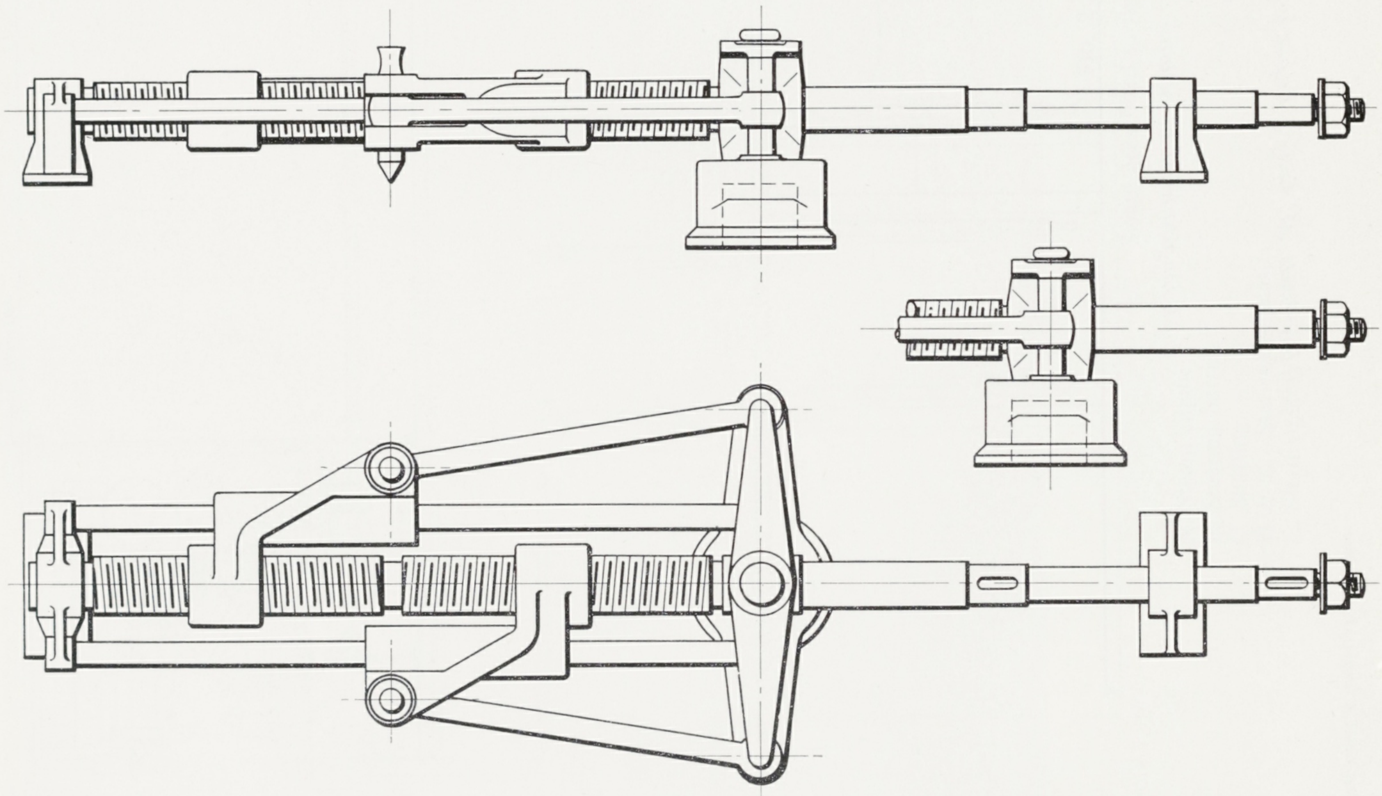


FIG. 1

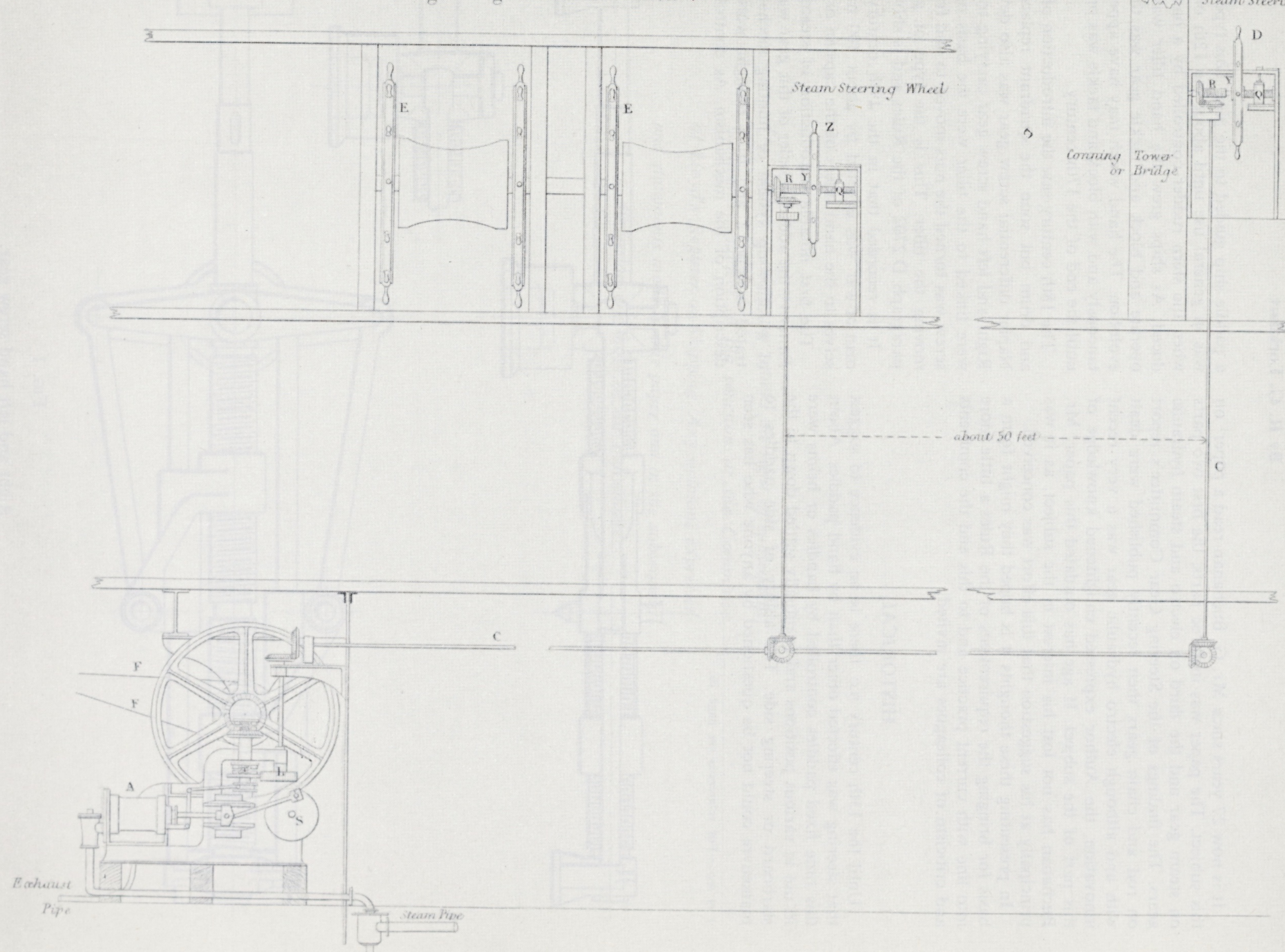
Right and left hand screw gear.

Trans. Inst. Naval Architects, Vol. 10, 1869

Plate V.

To illustrate Mr. Gray's Paper on Steam Steering Apparatus.

Fig: 1. Longitudinal Elevation.



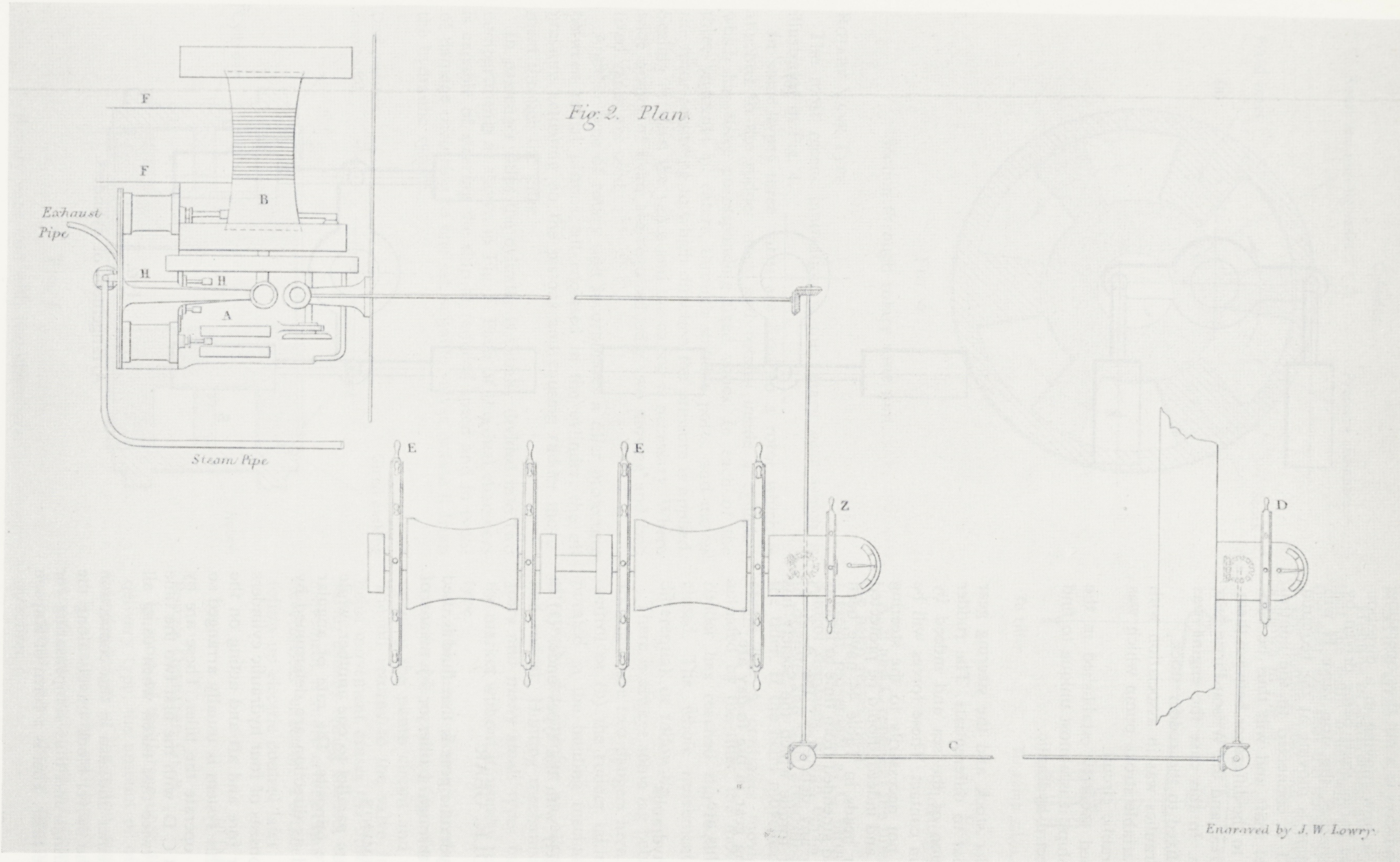


FIG. 2

Steam gear fitted to the *Great Eastern*.

A. Steam engine.
B. Steering barrel.
C. Shafting.
D & Z. Steering wheels.
E. Stern steering wheels.

F. Steering chains or ropes.
H. Automatic stop valve.
I. Differential screw.
L. Spur wheel.
S. Crankshaft.

the differential screw I travels endways in a nut thus operating the stop valve H. The engine is now moving in a direction corresponding to the screw. The barrel shaft being driven by a pinion on the engine crankshaft S, the stop valve H will continue to open until the engine is moving at the required speed. The differential screw I is connected to the control shaft C and also to the barrel shaft and will thus be controlled by both and correspond to the difference between them.

The wastage of steam was great and the Wilson-Pirrie gear was evolved to cut this down. In this case the engine was geared to a toothed quadrant fixed to the rudder stock.

The first application of hydraulics was in association with accumulators and it was the variable stroke pump which was responsible for the closed hydraulic circuit.

Steam hydraulic gear seemed securely established in the mid 1930s but to-day in new ships it is almost unique to find any other power gear than electro hydraulic.

While the rudder, the rudder stock and the steering gear compose the unit it is essential to differentiate. The rudder can be affected by every action of the sea and indeed by anything in which it may be in contact. These forces will be transmitted to the stock, but not necessarily to the steering gear as in most cases this can and indeed should be protected from such forces. The rudder heads in Table 44 have been built up over a long period, and amended from time to time as service experience has shown any deficiency; the A $\times$ D value is the only function of torque which the Society has thought fit to use.

The various types of steering gear in use to-day are:—

(1) Hydraulic, and under this are
    (a) Hand hydraulic.
    (b) Hand and power hydraulic
    (c) Power hydraulic
(2) Electric gear
(3) Rod and chain

It is proposed to deal largely with the types under (1) in this paper.

HYDRAULIC GEAR

The basis of all types of hydraulic gear is that fluid is used to operate the rudder stock through a tiller or by means of a rotary unit fixed to it.

Cylinder and Ram Arrangements

In Fig. 3(a) the cylinders are parallel to one another, while in Fig. 3(b) the cylinders are opposite. The rate of angular movement of the rudder and its direction are determined by the hydraulic pressure.

In Fig. 3(c) the principle consists of four hydraulic cylinders placed either athwartships or fore and aft and acting on the rudder by a pair of rams. This system is usually arranged so that three combinations can operate the tiller. These are by A and B, C and D, and A B C D, with the first two the time for operation will be about twice that taken when using all four cylinders.

The rams are coupled together and also in this connection is a bushed swivel block. The swivel block travels along the tiller arm turning in its bearings and thus compensating for the angular movement of the tiller. This is a form of Rapson slide.

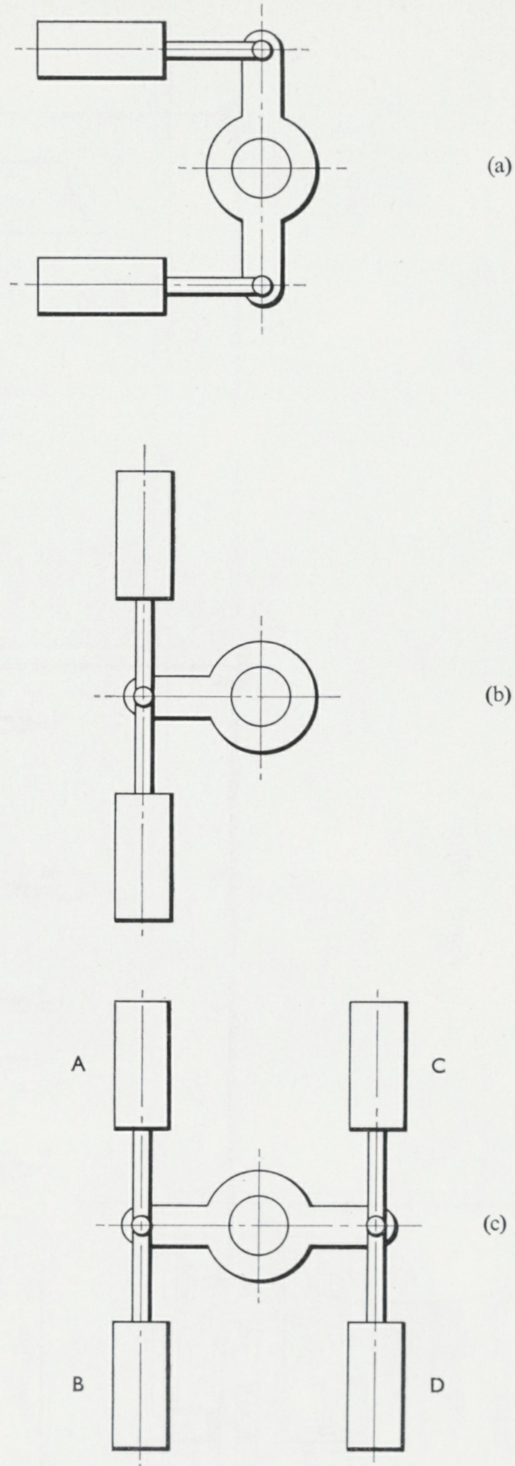


Fig. 3

Arrangement of cylinders.

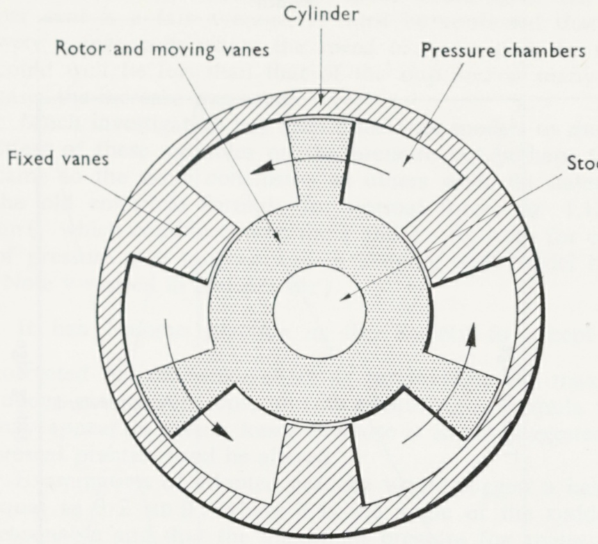
Cylinder


FIG. 4

Section through rotary vane unit.

ROTARY VANE TYPE

The most recent innovation is the rotary gear which is illustrated in Fig. 4.

In some types three vanes are fixed to a rotor which is attached to the rudder stock and rotates inside a cylinder, which has three corresponding fixed vanes. In each of the three segments oil enters, through suitable ports, and moves the rotor according to which direction the pressure is applied. Sealing between the vanes and the casing is necessary. Where wide angles of helm are required only two moving and two fixed vanes are used.

Another type of rotary unit incorporates a tiller projecting between radial pistons, all enclosed in the cylinder, the oil pressure actuating on the pistons and causing rudder movement through the tiller.

In passenger ships the cylinder is usually divided into two compartments as shown in Fig. 5. Each of the dual chambers is capable of steering the ship at reduced speed and in event of damage occurring to one set, suitable valves isolate it from the hydraulic system.

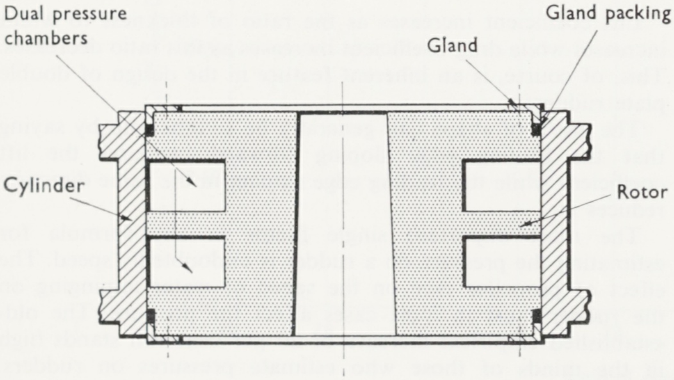


FIG. 5

Rotary vane unit with two chambers.

HUNTING GEAR

Hunting or differential gear is the mechanism which ensures that the helm order will be carried out without damage to the gear, and that the pump is cut off as required.

The hunting gear can be regarded as a floating lever and is shown in Fig. 6.

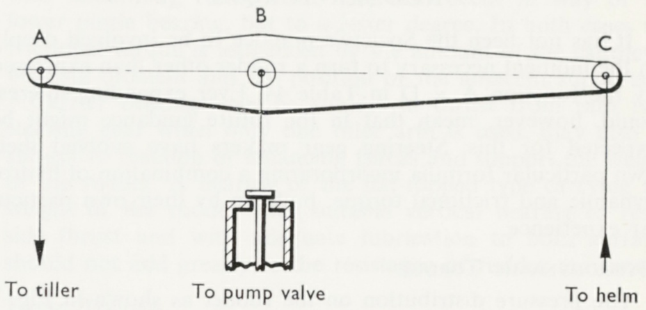


FIG. 6

Floating lever.

A, B and C represent the tiller, the pump stroke spindle and the helm respectively.

Helm movement at C through the telemotor or electric control displaces B and opens the valve, causing the pump to operate and thus press oil into the cylinder and operate the tiller. If the helm movement is now stopped the oil flow ceases but A continues moving in the opposite direction actuated by the tiller, the movement only ceasing when the rudder has reached the required angle and the valve has closed. The whole movement is referred to as hunting, differential, or follow-up.

There is always some lag between the helm and the rudder, and there are two dangers here, (a) the helmsman forcing the control or (b) the rudder taking charge. The storage incorporated in the hunting mechanism guards against these by effectively determining an allowable variance between helm and rudder. Hunting mechanism is a feature of most large gears and many small types but there are steering gears on the market without it, usually referred to as the non-follow-up type.

In this non-follow-up arrangement there is frequently a control valve, operated from the bridge and sometimes locally, opening the pump circuit and directing oil to the cylinders. The oil returns to the valve from the cylinders and as the helm movement ceases the flow is restrained and the control valve, usually spring loaded, cuts off the fluid pressure. The movement is more erratic than with hunting mechanism but nevertheless effective. Further reference is made to non-follow-up electric control later.

PUMPS

POWER OPERATED GEAR

In most of the larger units the pumps are of the variable delivery type, the amount of oil displaced is adjusted by the varying length of the piston stroke which causes displacement.

In smaller units and largely in the non-follow-up type the pumps are constant speed, constant output, rotary type and a control valve directs the pump output in the required direction.

HAND OPERATED HYDRAULIC GEAR

The steering column is provided with a built-in gear pump directly coupled to the steering wheel, the connections of the pump led through piping to the cylinder, the pump being operated by turning the hand wheel.

RUDDER TORQUE

It has not been the Society's practice to be involved deeply in the moment necessary to turn a rudder other than expressed by the function $A \times D$ in Table 44. Ever expanding interest could, however, mean that in the future guidance might be expected for this. Steering gear makers have evolved their own particular formula incorporating a combination of hydrodynamic and frictional torque, built up by their own particular experience.

HYDRODYNAMIC TORQUE

The pressure distribution on the rudder as shown in Fig. 7 is represented by the normal pressure, which can be resolved laterally by the lift and fore and aft by the drag. Each of these forces can be accepted as being inversely proportional to AV^2 where A is the area of the rudder in square feet and V is the speed of ship in knots and can thus be expressed as a coefficient of AV^2.

If the coefficients are accepted as C_L, C_D and C_N for lift drag and normal pressure respectively it can be accepted from Fig. 7 that

$C_N = C_L \cos \alpha + C_D \sin \alpha$ when α is the angle of attack.

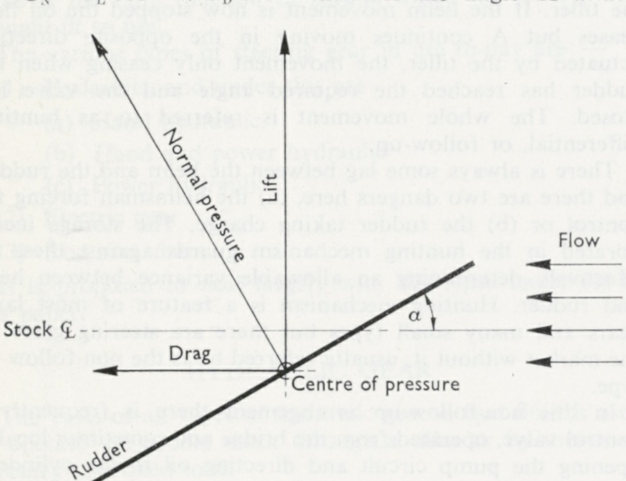


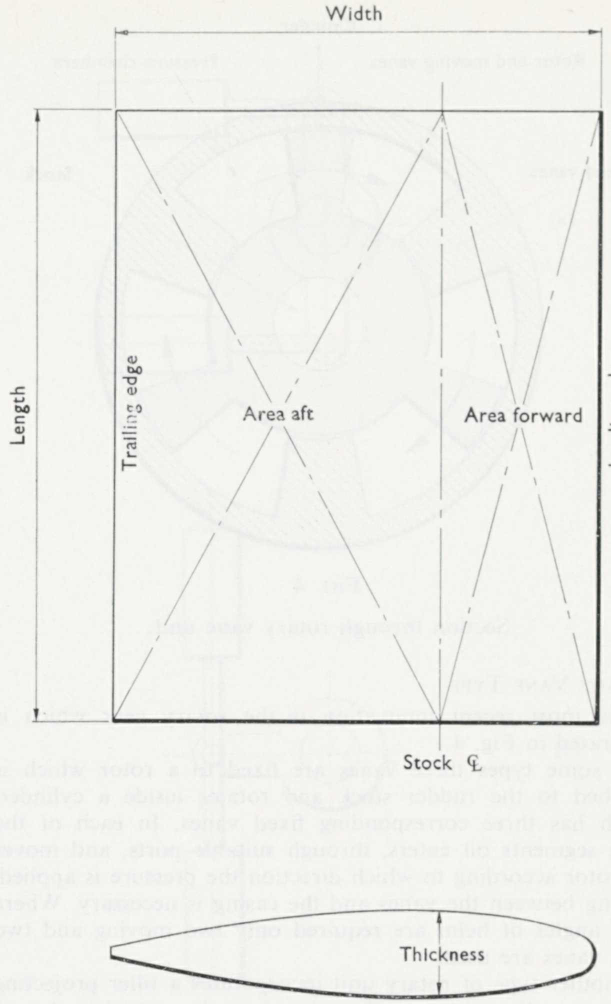
FIG. 7

Pressure distribution on rudder.

For simplicity of explanation of the terms that follow it is hoped that Fig. 8 will be of some assistance.

Each of these coefficients is affected by aspect ratio (i.e. length to width of rudder), by the ratio of thickness to width of and by the shape of the rudder.

The lift coefficient C_L is large at small rudder angles for large aspect ratios; at an aspect ratio of unity or 1:2 the lift coefficient will be maximum around 30°–35°. Drag coefficient C_D rises steadily to a maximum around 80° except in the case of aspect ratios around 1:2 where it can be high around 45°.



$$\text{Aspect ratio} = \frac{\text{length}}{\text{width}}$$

FIG. 8

Rudder outline for terminology.

Lift coefficient increases as the ratio of thickness to width increases while drag coefficient increases as this ratio decreases. This, of course, is an inherent feature in the design of double plate rudders.

The effect of shape can generally be summed up by saying that the trailing edge sloping forward increases the lift coefficient while the leading edge sloping in the same direction reduces it.

The most important single factor in any formula for estimating the pressure on a rudder is undoubtedly speed. The effect of propeller race on the speed of water impinging on the rudder must in most cases affect the pressure. The old-established empirical increase of 20 per cent still stands high in the minds of those who estimate pressures on rudders. In so far as propeller pitch $\times$ revs. represent a reasonable measure of propeller race it is difficult to find fault with this increase. An examination of this information extracted from

First Entries and related to the speed would agree that 15–20 per cent is a fair average. It must be confessed that there were a few cases where the speed of water past the rudder could well be less than that of the ship and in many other cases the increase was small.

Much investigation has been done with models to show the effect of these variables on the pressure but perhaps Gawn[3] came to the same conclusion as others when he stated that the old empirical formula for normal force, viz. 1.12 $Av^2 \sin\theta$ which combined with the $\frac{3}{8}$ and $\frac{1}{3}$ strip rules for centres of pressure give torques well in keeping with model results. (Note v=speed in feet per sec.)

It has become practice in the Society to accept $\frac{AV^2}{1120}$ corrected for the wake effect as representing the maximum rudder pressure in tons, and in advancing a formula which may appear to give a lesser pressure it is not suggested that present practice need be altered.

Examination of adopted practice would suggest a basic C_N equal to $3{\cdot}2 \sin\theta$, where θ is the angle of the rudder, as reasonable and that the maximum pressure for angles up to 35° for the rudder behind the screw can be given as $C_N \frac{A(Vw)^2}{1220}$ where $(Vw)^2 =$ the speed of the ship in knots increased for the wake effect, and A=the rudder area in square feet.

The centre of the pressure from the leading edge can be expressed as $({\cdot}165 + {\cdot}36 \sin\theta) \times W$ feet, where W=mean width of the rudder in feet and θ is the angle of the rudder.

From this it is possible to arrive at a reasonable value for the hydrodynamic torque.

Frictional Torque

The frictional torque is an association of pressures at bearings, pintles, and rudder carriers and the frictional coefficient pertaining to the particular arrangement. In simplex and multi-pintle rudders the frictional resistance should not be high; in two pintle rudders, or those having a pintle at the bottom and a bearing at the top, provided alignment is good and lubrication is sufficient the additional torque will be reasonable.

With spade rudders a considerable resistance to turning can accumulate from the bending forces on the lower bearing and with underhung rudders this can also occur in way of the lower pintle bearing, but to a lesser degree. In both cases the frictional torque is influenced by the amount of the cantilever bending moment and the position of the adjacent supports.

Carrier bearings can be subject to thrust from ram type steering gear when only one tiller arm is used, may also be subject to reaction or balancing forces and support the weight of the rudder. A bearing of the flat-topped type carrying the weight of the rudder with suitable vertical bearing to resist side thrust and with adequate lubrication to both surfaces should not add greatly to the resistance of rudder movement.

Total Torque

Since A × D is a fair measure of rudder shape it is suggested that torque could be expressed as

$$\frac{A \times D\ (V \times (1 + {\cdot}01\sqrt{L}))^2 C_t}{1220} \times \frac{1{\cdot}36}{C_b + {\cdot}68} \text{ ft. tons}$$

where A is the area of the rudder in square feet.

D is the centre of gravity of A from the centre of the pintles in feet.

V is the maximum sea speed in knots.

C is a coefficient as given in Fig. 9.

L is the length of the ship in feet.

C_b is the block coefficient at load draft.

The minimum value of $(1 + {\cdot}01\sqrt{L})$ should be 1·15. In twin screw ships having a rudder at the centre line some reduction of this torque value could be permitted. There is little doubt that in balanced rudders when the area forward of the stock exceeds 25 per cent of the rudder area, the torque becomes erratic and difficult to predict. The unusual rise in the value of C_t at ratios less than 4:1 is provided to take account of this.

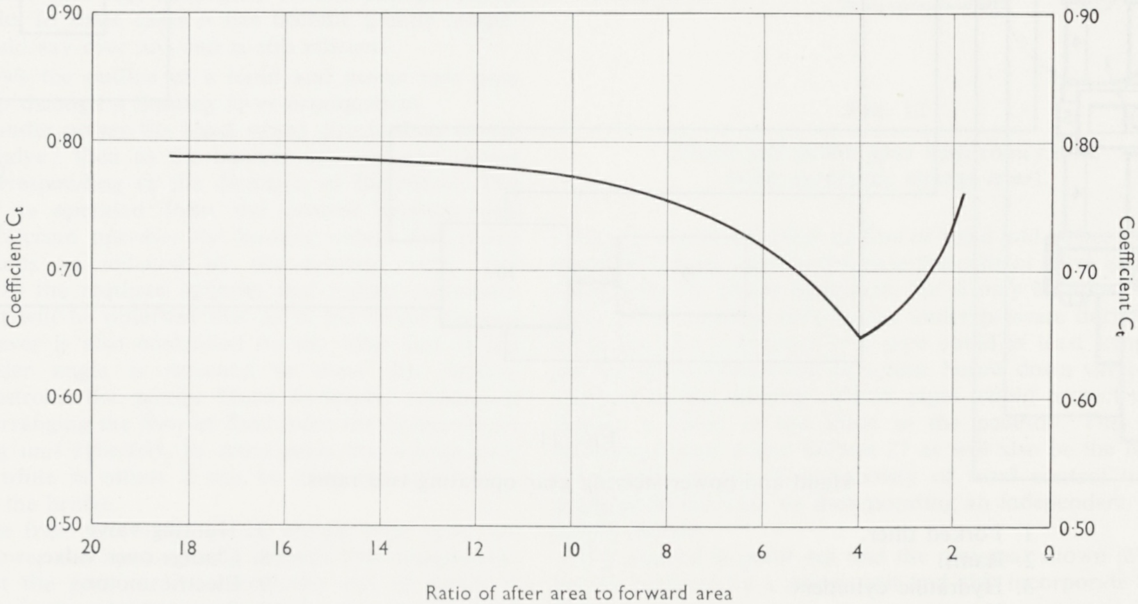


Fig. 9

Coefficient C_t values.

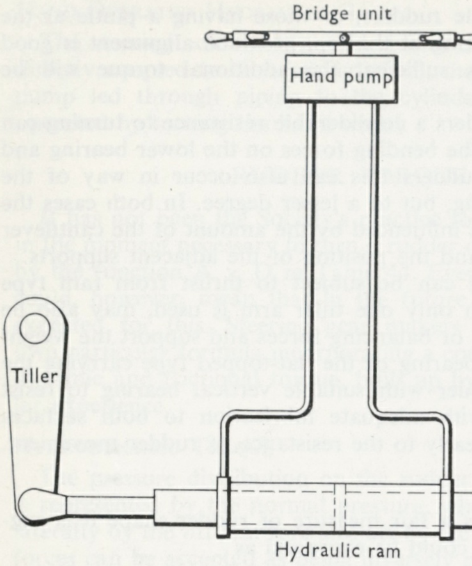


FIG. 10

Hand hydraulic gear for twin rudders.

In rudders of spade and underhung type a further increase to the formula is probably necessary to take account of the additional frictional torque in these cases.

This increase is given by the factor $(1+\cdot025\sqrt{d})$ for spade rudders where d is the maximum diameter of the stock in inches and by $(1+\cdot012\sqrt{d_p})$ for underhung rudders where d_p is the diameter of the lower pintle in inches.

INITIAL TURNING MOMENT

When the rudder is moved over, the initial turning moment of the ship is greatly influenced by the deadwood surrounding and adjacent to the rudder. Rudder posts and horns can probably increase the effective rudder area by up to 25 per cent without materially affecting the lift.

In the case of spade rudders and to some extent balanced rudders without back posts due to absence of deadwood the pressure on the rudder can be sensibly increased. This increase in pressure in the case of twin rudders behind twin propellers can cause an appreciable increase in torque at high speeds and Gawn[3] states that the pressure coefficient can be increased by over 20 per cent.

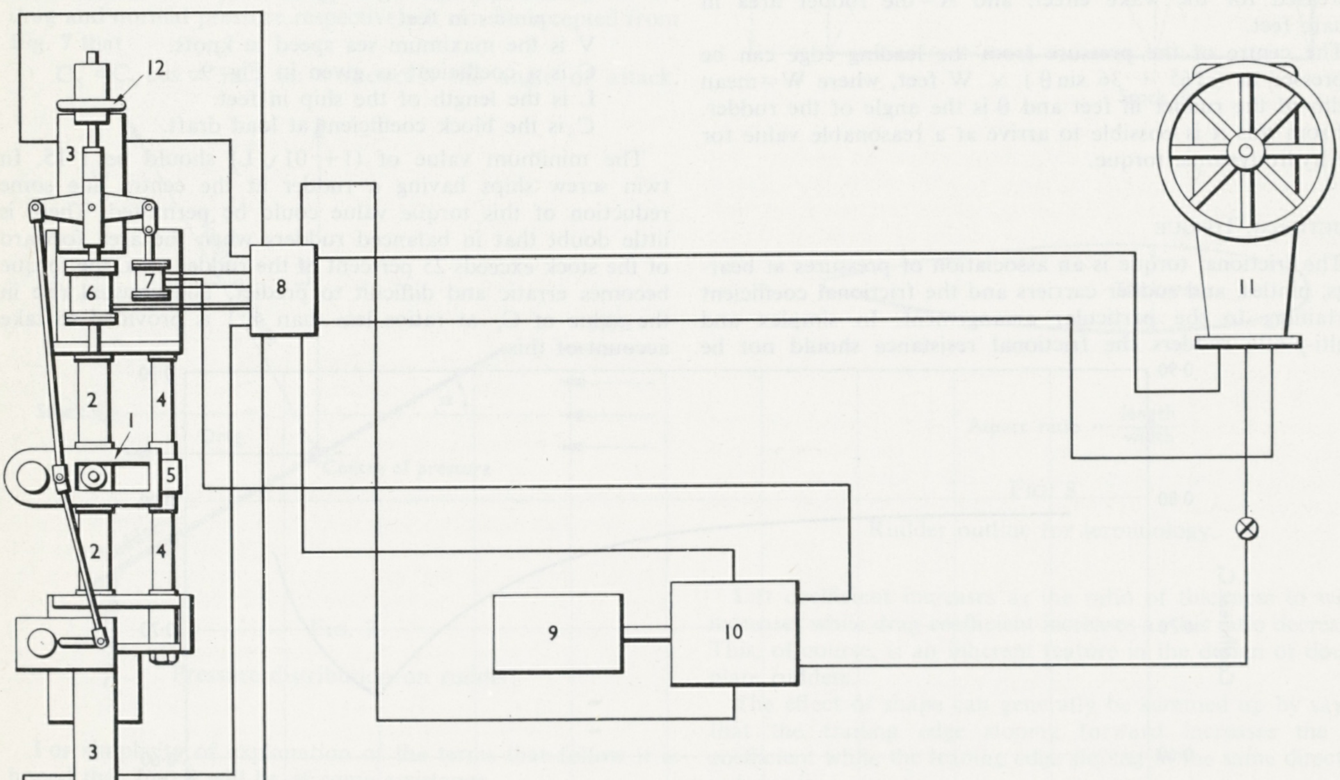


FIG. 11

Hand and power steering gear operating two rams.

1. Forked tiller.
2. Rams.
3. Hydraulic cylinders.
4. Tie bar.
5. Ram guide bracket.
6. Control cylinder.
7. Hunting valve.
8. Change-over valve.
9. Electric motor.
10. Power pump.
11. Bridge pedestal.
12. Local control.

TYPES OF GEAR

Hand Hydraulic Gear

Fig. 10 shows a diagrammatic outline of hand hydraulic gear operating twin rudders. The pressure is derived from the motion of the hand wheel which through suitable gearing operates a pump in the steering column and directs fluid to the rams. If it is desired that such a unit should operate the rudder or rudders from hard over to hard over in 30 seconds the maximum torque would not be much in excess of 3 ft./tons but if time is extended and sufficient effort applied this torque can be increased.

The alternative means of steering is provided by a separate tiller suitably secured to the stock and operated by hand or by block and tackle gear.

Hand and Power Gear

In this arrangement the power unit comprises the main gear and generally the hand steering the auxiliary gear. It is, however, necessary to discriminate between two particular systems of hand and power gear.

(a) The main gear is power hydraulic and the secondary gear is by turning a large hand wheel which, through a driving shaft and the necessary gearing and possibly a friction clutch, rotates the rudder. Such an arrangement of hand steering is shown in Fig. 16.

(b) The main gear is power hydraulic but with this arrangement the alternative means of steering is by hand hydraulic and can best be described as combined hand and power hydraulic gear.

It is easy for confusion to arise as to how far this gear complies with the Society's requirements. The hand wheel in the bridge can be the operator of both the power and the hand functions and in this respect is common. It is, however, prudent to remember that in one sense the hand wheel is acting as a controller to a power gear while in the other it is a direct connection to the hand pump in the pedestal and thus the source of power to the hand gear.

This particular gear has undergone radical change during the last decade. In some cases it has become greatly simplified, some could say over so; but is still efficient.

Fig. 11 shows the outline of a hand and power ram gear with follow-up through a floating lever arrangement.

Operating under power the hand wheel directs fluid to the change-over valve; then to the control cylinder and causes movement corresponding to the direction of the wheel. The floating lever is operated from the control cylinder and through the fulcrum operates the hunting valve. The pump circuit functions as induced by the hunting valve and delivers fluid to the required cylinder and rudder movement commences. It will be observed that as in the *Great Eastern* the floating lever is also controlled by the tiller and at the required rudder angle is returned to close the hunting valve and unstroke the pump. Hand hydraulic steering is obtained by arranging the flow of fluid from the bridge direct to the steering unit cylinders. In some gears the change over is automatic while in others it can be done manually by a lever valve at the bridge.

As the pipes from the bridge to the change-over valve are common to power and hand steering it needs little imagination to decide that the gear can be completely out of action if these pipes are fractured. However, in such an event assuming the power pump circuit is still functioning the gear can be operated by the hunting valve through the local control. In some cases the change-over valve is operated by electric control from the bridge.

Fig. 12 shows an outline of a hand and power hydraulic gear with rotary steering unit, it may be noted that no follow-up is incorporated in this gear.

Pipes from the steering column deliver oil to the control valve through an automatic change-over valve, the pump circuit thus being applied to the rotary unit and moving the rudder through the required angle as indicated on the steering column. In the event of a breakdown in the power circuit the automatic change-over valve directs the fluid to the cylinders and steering is then being effected by hand hydraulic operation. A local control lever is also provided.

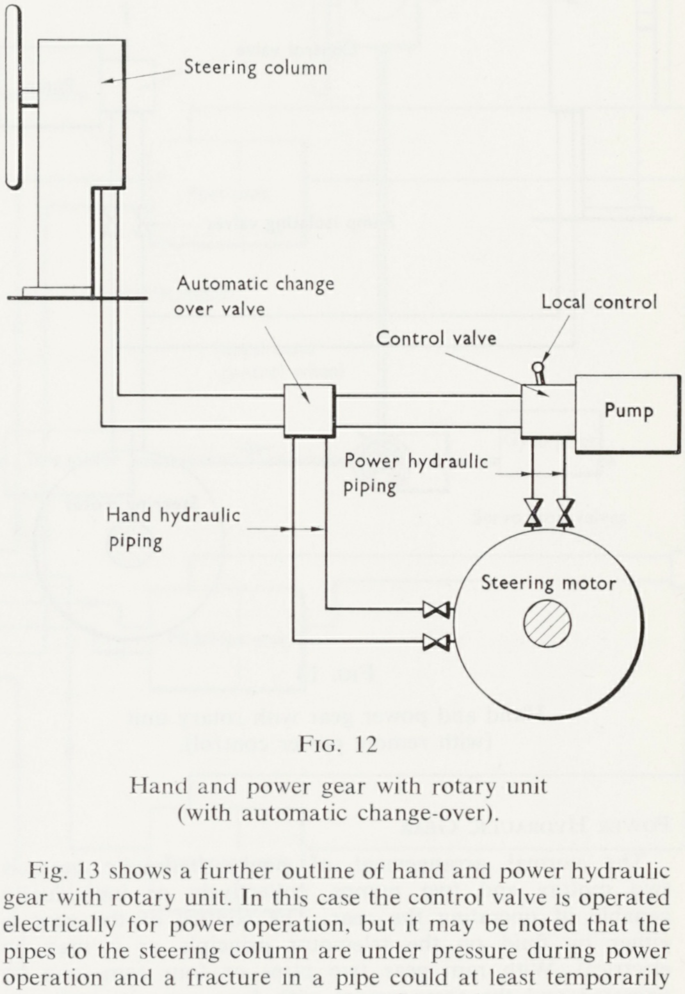


Fig. 12

Hand and power gear with rotary unit (with automatic change-over).

Fig. 13 shows a further outline of hand and power hydraulic gear with rotary unit. In this case the control valve is operated electrically for power operation, but it may be noted that the pipes to the steering column are under pressure during power operation and a fracture in a pipe could at least temporarily put the entire system out of action. Screw down valves at the junction of the steering column pipes would protect the gear against a break in the leads to the pedestal. This will be mentioned later under Section 27 as will also be the fitting of a hydraulic brake. The necessity of local control has been obviated in this case by incorporating an independent remote power control.

It is desired to point out that the ram gear shown in Fig. 11 can be replaced by a rotary unit and also incorporate hunting gear.

There are undoubtedly differences between the various types of hand and power gear, the Society can only subscribe

to a reasonable minimum requirement and it is the Owner's responsibility to decide the type of gear he will choose. It may be, however, that the description given together with the sketches will be of some use to colleagues in discussing this particular type of gear and help to resolve doubts relating to power and hand gear.

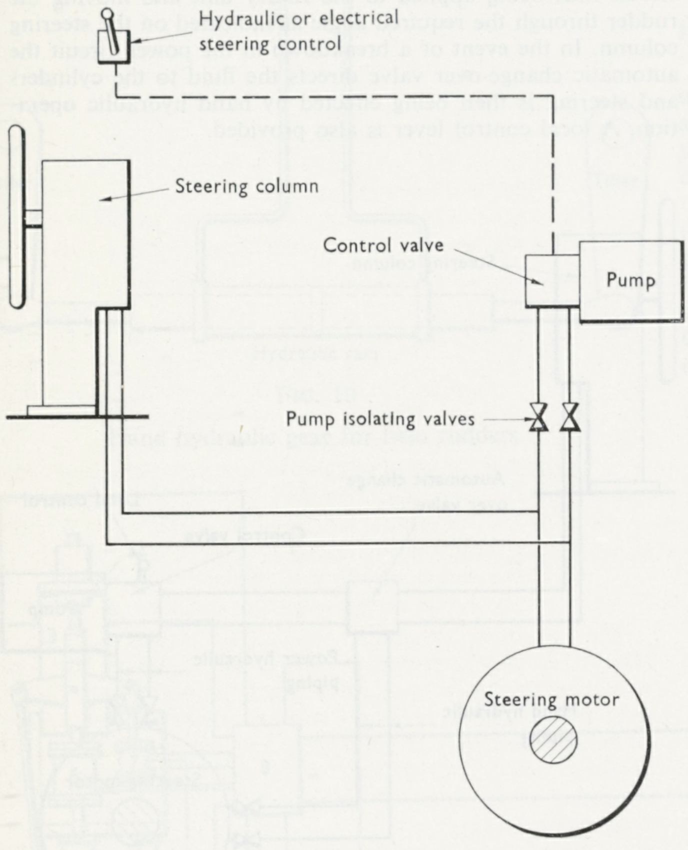


Fig. 13

Hand and power gear with rotary unit (with remote power control).

Power Hydraulic Gear

The normal arrangement of power hydraulic gear is two motors and two pumps, collectively or individually capable of operating the gear. The control of the gear is either by fluid on the telemotor principle or it may be electrical. With ram gear two rams or four rams can be operated and the arrangement with four rams is similar to Fig. 11 but with two or more pumps and motors in the circuit.

An interesting power ram unit[5] of this type has electro hydraulic as its main source of power but in the event of complete electric failure, a second power unit with the pump operated by steam or compressed air takes over the hydraulic operation automatically. On electric breakdown the solenoid is de-energised and this opens the valve of the steam or air engine, which then takes over as power operator. Some owners show a favourable leaning to this type of gear, in view of the fact that the secondary means of operation is obtained from a different source of power.

In large passenger ships it is quite usual to have four ram gear operated by three similar power units. Such an arrangement is shown in Fig. 14. When working one unit the time for hard over to hard over is about one minute, with two units this can be reduced to half a minute and with all three together the time is reduced to about 20 seconds. One unit can thus be considered as a standby for emergency.

As rotary steering gear is comparatively new, Fig. 15 shows an arrangement of power steering with this type of gear having duplicate pumps and motors and electric control.

In this gear with the follow-up arrangement, operation of the bridge steering column control operates one of the hunting motors in association with the rudder watcher and converter I. Follow-up is obtained through the return action of the floating lever on the rudder watcher and thus causing the hunting motor to stop.

Non-follow-up is obtained by push button control at the pedestal, this control directly operating the second hunting motor in association with converter II. Movement ceases only when the push button is released, the rudder angle being recorded on the bridge rudder indicator. In the event of breakdown in both controls the gear can function by the local hand control.

In some electric systems of steering control movement of the bridge operator produces electrical impulses. An amplifier or control unit transmits these impulses to the steering gear flat where an after power unit translates these impulses into mechanical movement of the power steering system.

Electric Control

Electric control of a power steering unit can be effected on the follow-up and non-follow-up systems, and also by automatic pilot.

Follow-up System

With this system the rudder movement will follow the hand wheel or controller and at the required angle movement will cease, the controller remaining in the position as already set.

Non-Follow-up System

In this case as the controller is moved from the neutral position the rudder will continue to move until the wheel or lever has been returned to the central position. Rudder movement can only be stopped with the controller in the neutral position or with the rudder against the stops. A rudder indicator is necessary to convey the rudder movement to the helmsman.

Automatic Pilot

Signals are received from the master compass and the ship is automatically held on a selected course. The rudder is set amidships but as soon as the compass indicates a deviation the automatic pilot causes sufficient helm movement to return to the selected course. It is usually possible to alter course by a few degrees through a trim switch and without introducing hand control. The automatic control is dependent on the feed back from the after transmitter to the feed back motor of the bridge unit.

All-Electric Steering Gear

The adaptation of the electric motor coupled through gearing to the rudder stock has perhaps been more popular on the other side of the Atlantic but has also been used to some extent in this country and the Continent. Fig. 16 shows an

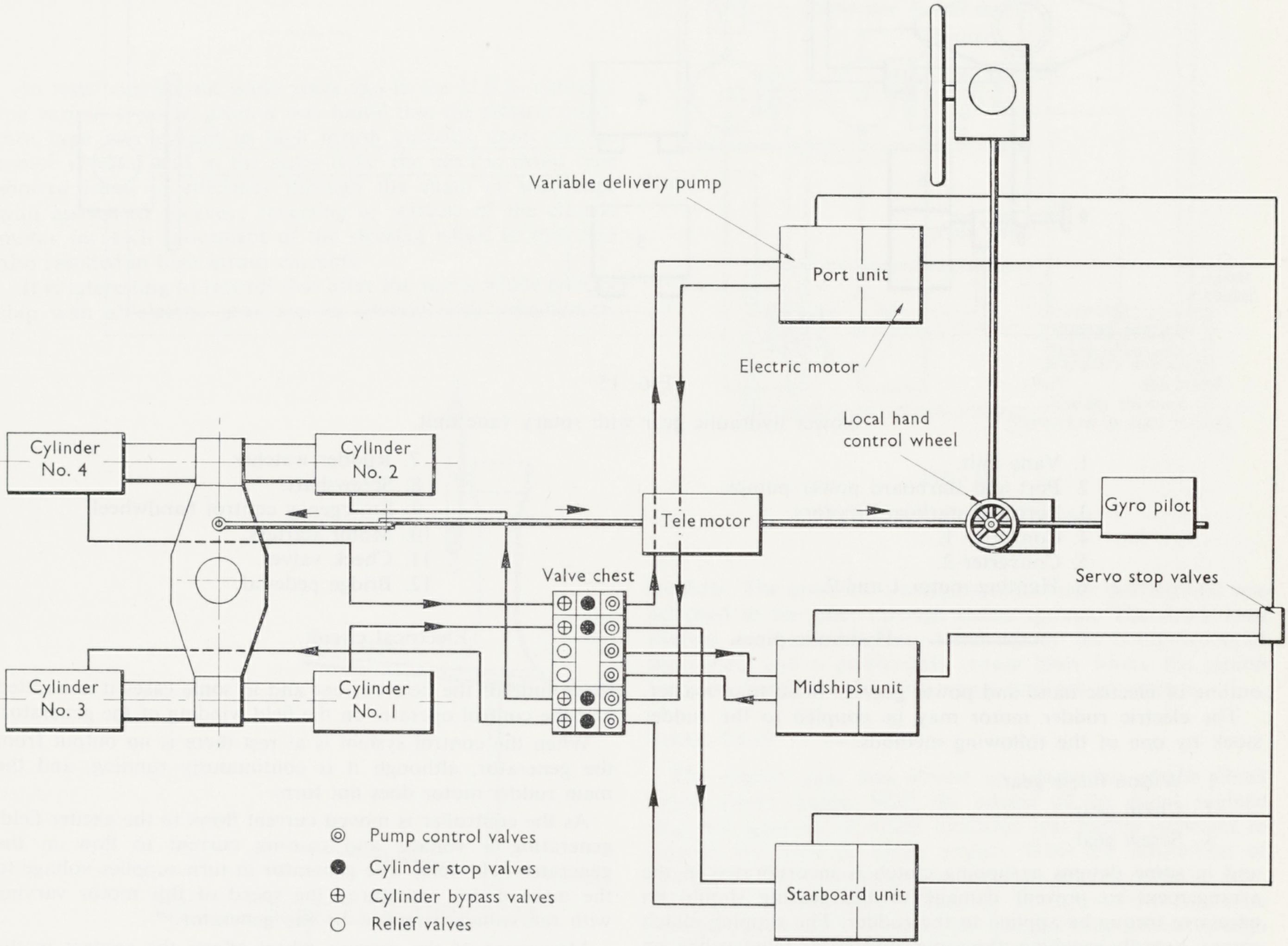


FIG. 14

Four ram gear with three pump units for large passenger ship.

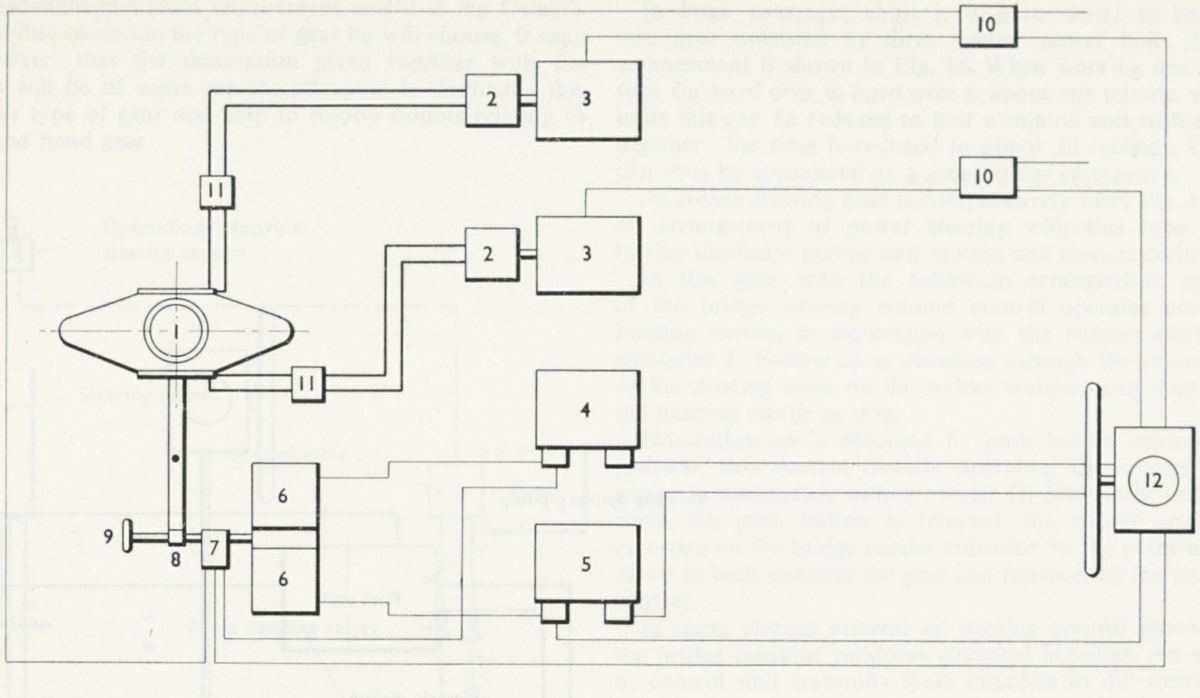


FIG. 15

Power hydraulic gear with rotary vane unit.

1. Vane unit.
2. Port and starboard power pumps.
3. Port and starboard motors.
4. Converter 1.
5. Converter 2.
6. Hunting motor 1 and 2.
7. Rudder watcher.
8. Screwshaft.
9. Emergency control handwheel.
10. Motor starters.
11. Check valves.
12. Bridge pedestal.

——— Hydraulic pipes. ——— Electrical circuit.

outline of electric hand and power gear as fitted to a coaster.

The electric rudder motor may be coupled to the rudder stock by one of the following methods:—

1. Wilson-Pirrie gear.
2. Chain gear.
3. Screw gear.

and in some designs a slipping clutch is incorporated in the arrangement to prevent damage to the gearing should an excessive torque be applied to the rudder. The slipping clutch serves, broadly speaking, the same purpose as relief valves on hydraulic gear.

There are two systems commonly employed in activating the rudder motor:—

1. Ward Leonard System.
2. Single Motor System.

1. Ward Leonard System

Fig. 17 shows a diagram of such a system. The gear consists of a motor generator continuously running at sea and having an exciter which provides the field current of the generator; the use of the exciter is largely determined by the magnitude of the field current and in some cases it is omitted and the control operates on the field winding of the generator.

When the control system is at rest there is no output from the generator, although it is continuously running, and the main rudder motor does not turn.

As the controller is moved current flows in the exciter field, generating a voltage and causing current to flow in the generator field coils. The generator in turn supplies voltage to the main motor armature; the speed of this motor varying with the voltage supplied by the generator.

Movement of the steering wheel offsets the contact in the wheel-house rheostat and a potential difference across the exciter field causes the main rudder motor to rotate. When the rudder has reached the required angle the contact on the rudder rheostat is coincident with the contact in the wheel-house rheostat, the steering motor stops, and the system comes to rest.

The voltage of the generator and, therefore, the speed of the steering motor depends on the difference in position between the bridge and rudder rheostat sliders, so that for large degrees of helm the movement starts off quickly but tends to ease as coincidence is approached.

The arrangement gives sensitive control, the motor starting smoothly, accelerating to a maximum and eventually stopping smoothly.

2. Single Motor System

The motor which runs only as the rudder moves, operates on the rudder head through gearing and generally through a quadrant (*see* Fig. 18).

Duplicate motors with duplicate feeders are normally fitted. The motor is usually slow speed series wound and an electro-magnetic brake prevents overspeeding.

Control in this case is effected by operation of the steering wheel which closes the control switch. The motor will run while the switch is held closed and stops only when the switch is opened.

In tests[4] carried out some years ago in the U.S.A. comparing various types of gears it was found that the electric quadrant type was subject to high inrush currents upon electric motor reversal and in the same tests[4] the electric drum type showed losses in efficiency through the chain or wire rope with associated sheaves; reversing or starting of the electric motor for each movement of the steering wheel in this case also resulted in high inrush currents.

It is interesting to record[5] that after the war a whale factory ship with all-electric gear was in contact with considerable ice floes. The quadrant was operated through gearing and was attached to the tiller through buffer springs. The shock load exerted on the rudder was taken up by the compression of the spring and a particularly severe blow broke the pinion from its attachment and the vessel was out of control.

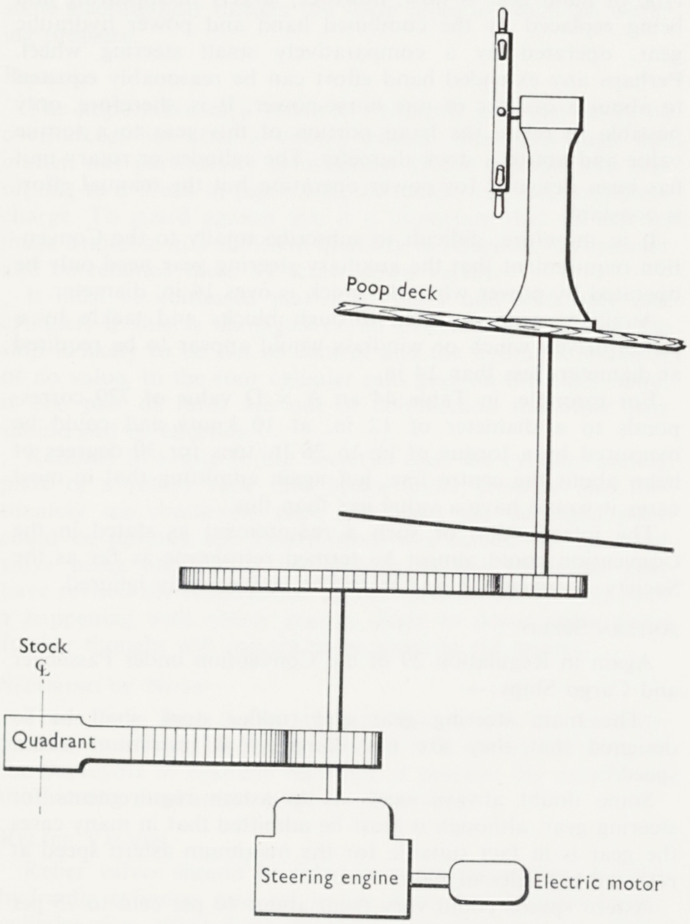


Fig. 16

Outline of electric and hand gear.

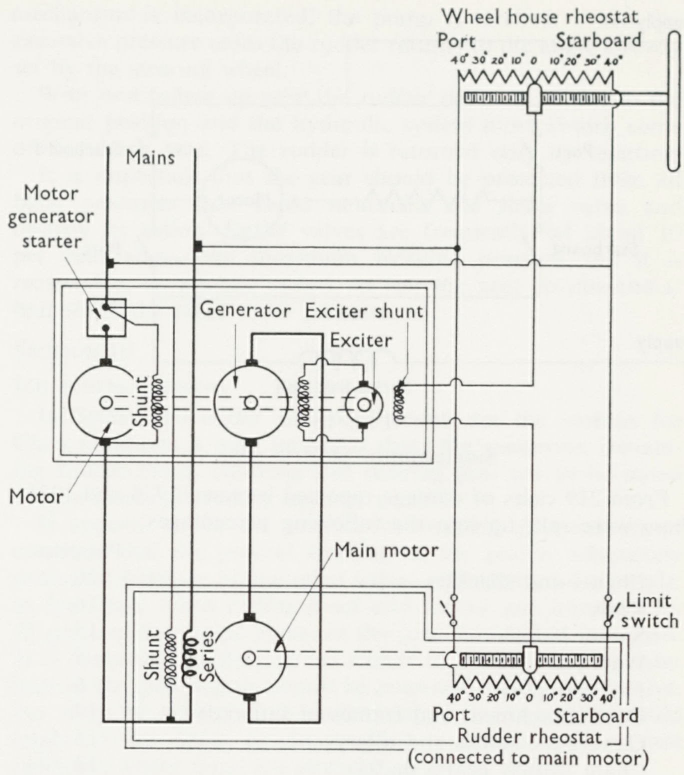


Fig. 17

Ward Leonard electric gear.

Steam Gear

The electric gear was almost an adaptation of the direct acting steam engine. With the advent of the diesel engined ship, the quantity of steam available was hardly sufficient to operate this type of steam engine. With the adaptation of hydraulics the movement was towards steam hydraulic gear. This gear is essentially the same as the electric hydraulic power gear already described, the pumps being driven by high speed steam engines. A valve connected to the pump control allows the engine to idle when not required to operate the rudder. The adaptation of a governor provides for complete control of the steam consumption. With the hydraulic gear the steam consumption was only about one-third of that required in the direct acting steam engine.

Rod and Chain Gear

In new ships rod and chain gear is now confined to small ships and very often to those in service in sheltered waters, and operated only by a hand wheel. It is thought worth-while giving the findings of the Steering Gear Committee published in 1936 as to the vulnerability of the different parts of the gear.

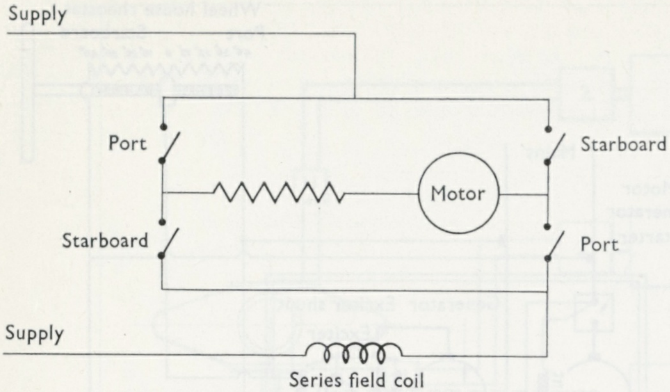


Fig. 18

Single motor electric gear.

From 219 cases of damage reported between 1926 and 1935 these were split up into the following percentages:—

	Per cent
Chains and shackles	26
Rods	6
Spring buffers	15
Warwick screws	5
Fairlead pins	6
Deck attachment and frames of fairleads ...	10
Quadrants, stocks and tillers	6
Chain drums, gears, shafts	12
Miscellaneous	14

It would be interesting to hear any comments from colleagues who have surveyed rod and chain gear mechanism with power unit within recent years.

The source of the requirements of D 2710 relating to the spare gear of sea-going ships of 12 knots and over was found in the ultimate paragraph of a letter from the Chamber of Shipping dated 27th April, 1937, stating:—

"In ordering new ships shipowners who intend to install rod and chain steering gear should be careful to specify in particular that gears should not be less in strength than that laid down by Lloyd's Register for a 12-knot ship, that detailed plans should be submitted to the Classification Society concerned at the earliest possible moment and that the advice of the Classification Society as to the strength of components and the other matter dealt with above should be carried out."

MAIN AND AUXILIARY GEAR

In dealing with a rudder moving to hard over in any specific time, it would be only reasonable to state the degree of hard over anticipated. In fixing the time limit, it must be appreciated that in the design of hunting mechanism a certain slowing up takes place at the end of the pump stroke before the rudder comes to the ultimate angle at the hard over, and account should be taken of this in the time requirement.

The 1960 Safety Convention (Regulation 29) Passenger and Cargo Ships states:—

"The auxiliary steering gear shall be of adequate strength and sufficient to steer the ship at navigable speed and capable of being brought speedily into action in an emergency", and in the same section under Passenger Ships only states:—

"The auxiliary gear shall be operated by power in any case in which the Administration would require a rudder stock of over 9 in. diameter in way of the tiller", and further for Cargo Ships states:—

"The auxiliary steering gear shall be operated by power in any case in which the Administration would require a rudder stock of over 14 in. diameter in way of the tiller."

Auxiliary Gear

The auxiliary or secondary means of steering has never been expected to undertake the complete duty of the main gear. It would, however, be reasonable to expect the secondary gear to turn the rudder through at least 20 degrees either side, although a measurement of the time need not be related to this particular angle of helm.

Ten knots or two-thirds of the maximum ahead speed, whichever is less, could be accepted as a navigable speed. The Society's practice is to accept hand gears as alternative means of steering for cargo ships up to a rudder stock diameter of 10 in. at a speed of 10 knots. Table 44 shows that this corresponds to an A × D value of 450. An examination of this numeral value would suggest that for an angle of 15 degrees on one side to 15 degrees on the other the maximum torque would not be likely to exceed 15 ft./tons and in most cases would be less.

It is almost certain that the large steering wheel with the necessary manual effort is the only type of hand gear likely to operate a rudder requiring a 10 in. stock at 10 knots. This type of hand gear is now, however, largely disappearing and being replaced by the combined hand and power hydraulic gear, operated by a comparatively small steering wheel. Perhaps any extended hand effort can be reasonably equated to about a quarter of one horse-power. It is, therefore, only possible to relate the hand portion of this gear to a torque value and not to a stock diameter. The cylinder or rotary unit has been designed for power operation but the manual effort is constant.

It is, therefore, difficult to subscribe totally to the Convention requirement that the auxiliary steering gear need only be operated by power when the stock is over 14 in. diameter.

Auxiliary gear operated through blocks and tackle by a power-driven winch or windlass would appear to be required at diameters less than 14 in.

For example, in Table 44 an A × D value of 720 corresponds to a diameter of 12 in. at 10 knots and could be measured by a torque of up to 26 ft./tons for 30 degrees of helm about the centre line, but again admitting that in most cases it would have a value less than this.

The introduction of such a requirement as stated in the Convention could almost be termed retrograde as far as the Society is concerned and might be conveniently ignored.

Astern Speed

Again in Regulation 29 of the Convention under Passenger and Cargo Ships:—

"The main steering gear and rudder stock shall be so designed that they are not damaged at maximum astern speed."

Some doubt always exists as to astern requirements for steering gear, although it must be admitted that in many cases the gear is in fact suitable for the maximum astern speed at reasonable angles of helm.

Astern speeds could vary from about 40 per cent to 75 per cent of the ahead speed. At the higher values the steering gear and the rudder stock would require special consideration. It is significant that the Convention does not specify any rudder

angle in association with the maximum astern speed. If the rudder is required to move to hard over going astern the ship would most likely be moving very slowly and such a condition would be likely to be an emergency. It is the practice of some steering gear makers to make reasonable provision for astern conditions, whilst apparently others are less concerned.

The Society is always prepared to examine the scantlings of the rudder and stock for any particular astern speed the owner may require and an example of this was dealt with in Mr. Last's paper[6] on Rudders and Sternframes.

GENERAL REQUIREMENTS

Piping

In combined hand and power hydraulic gear, where much of the piping is common to power and hand operation every precaution should be taken to isolate the systems so that failure of either would not be likely to result in inability to steer the ship.

Testing

Some doubt exists in the minds of Surveyors as to exact requirements of testing the component parts of the gear. These should be subject to a test of not less than one-and-a-half times the relief valve pressure. The testing of piping to at least this requirement is particularly important in hand and power gear where the pipes are common to both, and some steering gear makers test these pipes to twice the working pressure.

Brakes

The importance of a brake in hydraulic gear can easily be overlooked, but after a serious mishap some years ago this matter was particularly investigated. If the cylinder can lose oil due to a break in pipes, etc., the rudder can take complete charge. To guard against this it is important that oil can be made available in a sufficient quantity to replenish the cylinder and be retained there by screw down valves.

It must be confessed that in a rotary unit with only one cylinder, if this is damaged or an external seal broken the ship is likely to be out of control and the hydraulic brake is of no value. In the four cylinder ram gear on the other hand, if one pair of rams are out of commission the other pair should still be effective.

Recently in a ship to the Society's class the cylinder sealing plate of a rotary vane unit broke under pressure and fortunately sea conditions were good and the damaged plate could be patched up. This mishap was caused by the relief valve being neutralised, the makers of this particular gear have revised the construction of the valve in question. If such a happening with rotary gear is likely to occur again some further thought will require to be given to the matter.

Securing of Nuts

There have been a few mishaps due to nuts in the components not being secured and while it is hardly the duty of the Surveyors to examine each nut, if possible the security of the hunting mechanism might be examined.

Relief Valves

Relief valves should be fitted between the pumps and the hydraulic cylinders and set to lift when the pressure in the cylinder rises either from the results of abnormal sea conditions or direct pressure. The lifting of the relief valves allows the oil to by-pass causing tiller movement and when hunting mechanism is incorporated, the pump is stroked. When the excessive pressure eases the rudder returns to the angle already set by the steering wheel.

With non-follow-up gear the rudder does not return to the original position and the hydraulic system must absorb some of the shock load. The rudder is returned only by resetting.

It is important that the gear should be protected from all back pressures that could neutralise the relief valve and destroy its action. Relief valves are frequently set about 10 per cent above the maximum working pressure and it is reasonable, as already stated, to test the gear to one-and-a-half times the relief valve pressure.

Section 10

Ice Strengthening

In Section 10 under the requirements for the various Ice Class notations it may be noted that "the gudgeons, remaining rudder items, coupling and steering gear are to be based on the increased rudder head or pintle".

It appears to be the practice of steering gear makers to conclude that the general working of the gear is adequately protected from ice by the relief valves and where, for example, in Ice Class 1 the rudder head and pintles are increased by 25 per cent the actual power of the gear may not be increased. It is, however, the duty of the maker to design any particular part of the gear which cannot be protected by the relief valve, but may be subjected to undetermined loading. This could apply to the vanes of rotary gear or to the ends of ram cylinders where these are intended to act as stops.

STEERING GEAR DEFECTS FROM REPORTS

From examination of Reports on Steering Gear troubles in some 200 ships built since 1950 about 5 per cent of the defects were serious and resulted in grounding or the ship being out of control. The faults in this category were a mixture of mechanical and electrical and showed no consistency.

Under the various types of gears it is thought it might be interesting to mention the defects generally.

Steam Hydraulic

Crankshaft and bearings in the engine, resulting in extreme cases in breakdown of the gear.

Electric Hydraulic

Fractured and burned cables and the electric motor armatures burning out.

All Hydraulic Gear

Pump failure complete and partial. Telemotor pipes fractured, one such case, attributed to vibration, resulting in grounding; leaks of fluid generally of a minor nature; fractured rams and cylinders; bushes in the trunnion falling out, this was of some consequence in several cases; fractures in the hunting mechanism. Gear jamming from a variety of causes, but dirt in the fluid was often suspected.

Electric Gear

Quadrant teeth badly worn, this was serious in a few cases, motor and generator armatures burning out and cables fractured or burned.

General

In some of the groundings it is difficult to ascertain whether all the possible alternative means of steering were exploited.

It is a tribute to the steering gear makers in general that few cases were found of the gear known to be underpowered, although friction in bearings and carriers caused serious scoring, in some cases offering serious frictional resistance and illustrating the necessity of careful alignment. In general particularly in powered gears the time for hard over was less than the 30 seconds required by the Rules.

PARAGRAPHS D 2701–2705

It is suggested that existing paragraphs 2701–2705 could be revised where required along the following lines:—

(a) All ships are to be provided with two independent means of steering. One of the gears is to be capable of operating the rudder from 35 degrees over to 35 degrees over when running ahead. In ships exceeding 200 ft. in length this gear is to be operated by power, and should put the rudder from 35 degrees on either side to 30 degrees on the other side in 28 seconds when the ship is going ahead at full sea speed. In ships less than 200 ft. where the gear is power operated this requirement is also to be complied with.

(b) The secondary means of steering is to be capable of putting the rudder from at least 20 degrees on one side to 20 degrees on the other and when the diameter of the rudder head from Table 44 exceeds 10 in. (9 in. in passenger ships) at a speed of 10 knots is to be power operated and should put the rudder from 15 degrees on one side to 15 degrees on the other in 60 seconds when going ahead at 10 knots or two-thirds of the maximum ahead speed whichever is less.

(c) Where hand and power hydraulic gear form both means of steering and is operated through common pipes, the gear is to be capable of being controlled locally. Provision is to be made so that failure of the system between the steering wheel and the pump unit will not result in inability to steer the ship. The pump unit is to be arranged as near the tiller as possible.

TILLERS AND QUADRANTS

With hydraulic gear apparently firmly established the Society's requirements for tillers and quadrants although adaptable to current requirements could perhaps be revised.

It is thought that a formula would fit the general requirements for tillers and could be expressed as:—

The section modulus of quadrants and tillers clear of the boss to be not less than $\cdot 15d^3 \dfrac{(a-b)}{a}$ inches^3

where d = diameter of the rudder head in inches as required by Table 44.

a = the distance from the point of application of the load to the centre of the rudder stock in inches.

b = the distance from the boss to the centre of the rudder stock in inches.

The modulus at the end of the tiller could be about 33 per cent of this value.

The breadth to depth ratio of solid tillers should probably be restricted to a maximum of 2 : 1.

The outside diameter of the boss is generally $1\cdot 8 \times$ the diameter of the rudder stock and the depth equal to the diameter of the rudder stock.

Where more than one arm is fitted the combined modulus should, of course, be equivalent.

For quadrants and tillers used only as the alternative means of steering in sea-going ships the diameter of the rudder head in inches to be used could be based on the rudder numeral $A \times D$ in Table 44 for a speed of 10 knots.

The modulus at the boss could then be $\cdot 12d^3 \dfrac{(a-b)}{a}$.

Where a hand tiller can be fitted the modulus of the tiller could be proportional to the effort applied.

BOLTED TILLERS AND QUADRANTS

Tillers and quadrants are fixed to the head by bolting as required by Table 50 but it may be doubtful whether the distance between the bolt centres in the Table is necessary. The clamping can be achieved by bolts closer spaced provided their diameter is related to the stock, and possibly the diameter of the bolts in inches can be given as $\dfrac{\cdot 60d}{\sqrt{n}}$

when d = diameter of the rudder head in inches.

n = total number of the bolts.

RODS AND CHAINS

Table 49 is now only applicable to smaller ships and even in these ships it is almost a novelty to find rod and chain gears.

The diameters of rods and chains are, however, sensibly satisfied by the formulæ given by:—

The diameter of the chain in inches $dc = \sqrt{\dfrac{d^2}{R}}$ or $\dfrac{d}{\sqrt{R}}$

and the diameter of the rods in inches $dr = 1\cdot 12\sqrt{\dfrac{d^2}{R}}$

where d = diameter of the rudder head in inches as required by Table 44.

R = radius of the quadrant or length of the tiller at the centre of the chain in inches.

ACKNOWLEDGMENTS

I would like to thank the steering gear makers for the co-operation given, also for the assistance of colleagues in the preparation of this paper. I am also indebted to the R.I.N.A. for permission to reprint Fig. 2, and the relevant description.

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ABBREVIATIONS

R.I.N.A.—Transactions of the Royal Institute of Naval Architects.

L.R.S.A.—Lloyd's Register Staff Association.

S.N.A.M.E.—Society of Naval Architects and Marine Engineers.

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Discussion

on

Mr. R. G. Lockhart's Paper

STEERING GEAR

LLOYD'S REGISTER OF SHIPPING

71, Fenchurch Street, LONDON, E.C.3

Discussion on Mr. R. G. Lockhart's Paper

STEERING GEAR

Mr. O. M. Clemmetson

If Surveyors who are approached to write a paper for the Staff Association have a difficulty in finding a subject then they may take their example from the Author who has used a former title and brought the information up to date with what I am sure will have beneficial effects.

As regards the rotary vane steering gear, in case some should consider this a rather complicated gear which is not as substantial as the more conventional gears, I might mention that the strength of the vanes and their attachments is related to the torque required to cause the stock to yield plus a good margin, and the relief valve is set to blow at about one-sixth of the pressure which would cause the stock to yield so that there is a substantial safety factor.

On page 7, first column, I would suggest some clarification of the formula given in the fourth paragraph to enable it to be compared more easily with the Society's formula in the third paragraph. If the latter is expressed in terms of Vw and the usual 20 per cent wake effect used, then the formula for pressure becomes $\frac{A(Vw)^2}{778}$ and this can be compared with the Author's suggestion for pressure at an angle of 35°, i.e. $\frac{A(Vw)^2}{668}$ a reduction of about 13 per cent.

Regarding the expression for total torque, it would appear that C_t represents some proportion of the breadth D which takes into account the position of an assumed centre of pressure in relation to the turning axis and that this incorporates a frictional allowance. When the ratio of areas forward to aft of the factors is 4:1, i.e. 20 per cent of the area is forward, C_t is ·65 per figure 9 corresponding to ·85D from the leading edge, and when the ratio is 15:1, i.e. only 6 per cent of the area is forward, C_t is about ·78 corresponding to ·84D from the leading edge. In both cases, therefore, the assumed centre of pressure is in approximately the same position which is reasonable, but this position differs considerably from the usual $\frac{3}{8}$ or $\frac{1}{3}$ of the breadth usually adopted for streamline rudders and represents an increase in torque for friction of at least $2\frac{1}{2}$ times the hydrodynamic torque.

The expression for speed in the total torque formula gives an increase of 20 per cent for the wake factor (the Society's usual practice) in a 400 ft. ship, and the minimum value of 15 per cent occurs at 225 ft. and I should be interested to know the reason for this difference. Also the reason why a lower torque would be required for a fuller ship in accordance with the final part of the formula since it would appear that with greater displacement more inertia would require to be overcome.

Mention of the astern speed on page 14 draws attention to the fact that with Kort Nozzle rudders it is possible for high astern speeds to be obtained without instability in course keeping, and the maximum torque on these rudders when fitted with fins (as is usually the case) is in the astern condition.

On page 16 in the suggested amendments to the Rules, I would suggest that an absolute minimum speed in (b) should be seven knots in order to maintain way against an adverse tide.

Under tillers and quadrants the suggested modulus for tillers clear of the boss appears to give a stress on the tiller about 30 per cent greater than that on the corresponding stock which is reasonable, since in an emergency it would allow the tiller to distort before the stock. However, the formula is based on dimensions with the tiller amidships whereas the maximum torque occurs with the rudder and tiller at 35°. If the distance "a" is measured at the maximum angle a considerable difference in the modulus of the tiller could occur. For example, with a 9 in. stock dimension "b" would be 8·1 in. and dimension "a" with tiller amidships might be 24 in., but at 35° it would be 29·3 in. and this would represent an increase in $\frac{a-b}{a}$ of 10 per cent compared with the midship position. In such a case, therefore, the stress on the tiller with dimensions obtained from the midship position would be 44 per cent higher than the stock, which seems rather high.

The Author's comments on the International Convention requirements for steering gear are very pertinent, and it is to be hoped that they will bear some fruit.

Mr. J. Stevenson

The following points may be of interest in connection with the use of the Napier screw gear for auxiliary hand steering gear, Fig. 1. On one occasion when carrying out the test of this type of gear in still water it operated very satisfactorily. The Captain, who was present at the time, stated that in his experience the weak point in the gear was the attachment of the end support frames to the deck and this had failed on the one occasion when he had to use the gear. While the gear is man enough for the job at moderate speeds, it will be appreciated that the gear does not incorporate shock absorbing arrangements and, therefore, although some such arrangement can be devised, any blow on the rudder is transmitted direct to the gear causing complicated loading of the holding down bolts due to the height of the gear above the deck.

It is suggested that, where this type of auxiliary gear is proposed, the end supports and their attachment to the deck be considered in the light of resistance to the blows on the rudder transmitted to the tie rods.

Note also that the action of the operating screw and tie rods induces axial travel of the shaft, and clearance must be left for this between the end of the screw and the support bearings. The dimensions of the gear being known, the axial travel of the screw can be calculated.

The writer had experience of such a gear jamming after being overhauled. Fortunately, he was also aware of the possible cause and found that the end clearance had been mistaken for wear and new bearing bushes had been fitted to take up the slack. It did not take long to have the inner faces of the bushes machined to give the necessary clearance, after which the gear operated without further trouble.

Turning to page 4, Fig. 3(b), I should like to stress the importance of ensuring that the stroke limiting arrangements are set to safe limits before swinging the gear hard over. On

one occasion in answer to an urgent call, a visit was paid to a small passenger ship, when it was found that, in preparing for departure, the steering gear, which had been overhauled at the previous port, was being tested and went so far over that the swivel block put a bending moment on the starboard cast iron ram which snapped off where nipped in the cylinder gland. No cast iron was available so a steel shaft was turned to suit, installed and operated satisfactorily. The importance of correct adjustment of the travel limit controls was specially emphasised in view of the material of the new ram.

In conclusion, I should like to thank the Author for a helpful and interesting paper.

Mr. J. Wormald

This instructive and comprehensive paper will fill a long-felt want among outport Surveyors and our thanks are due to Mr. Lockhart for having responded to the encouragement which he acknowledges in his paper ; in addition he is to be congratulated on having provided "Buchanan's Steering Gear" with such an efficient and really modern follow-up.

This is the sort of back-room material which is invaluable to those who work outside and, whilst the paper itself offers little scope for comment or criticism, the following remarks will possibly give Mr. Lockhart an opportunity of enlarging upon one or two of the points he has mentioned.

Under the heading "Hunting Gear", Mr. Lockhart refers to the danger of the helmsman forcing the control and, later, under the heading "Steering Gear Defects from Reports (General)" he says that it is a tribute to the steering gear makers that few cases have been found of underpowered gear. Would Mr. Lockhart care to say whether it ever became apparent that it was possible for a very energetic helmsman to blow the motor fuses in one of the earlier type four-ram two-motor electro-hydraulic gears and thereby put the power gear temporarily out of action?

The earlier rules merely required that the two motors be supplied by two independent circuits and that each circuit be connected to the main switchboard but in 1961 the Rules were redrafted and it then became a requirement that "only short circuit protection and an overload alarm shall be provided on steering gear circuits". It would be interesting to know whether there is any connection between the previous question and the fitting of an overload alarm, as distinct from a fuse or a circuit breaker.

Whilst on this subject, have we not recently had substantiation of Mr. Lockhart's veiled advocacy of employing a secondary means of operation which is obtained from a different source of power? This reference is to the case of a large tanker which rammed a bridge with disastrous results because "a generator failure jammed her electric steering" (*vide* "Lloyd's List").

Mr. Lockhart's suggestion for the inclusion of minimum angles of rudder movement in paragraph 2701 is a surprising one and appears to be a retrograde step. Rudders and steering gears are much more efficient than they used to be and the number of occasions on which a ship moving at full speed in open waters, or at moderate speeds in closer waters, requires to put the rudder over 30° should be negligible. If, however, it is a question of improving the manœuvrability of a ship moving slowly in restricted space, or under adverse weather conditions, would it not be better to legislate for the adoption of up-to-date and more effective methods of achieving this object rather than make a requirement of something which has been more or less standard practice for years?

The suggested revision of the requirements for tillers and quadrants will, presumably, have the effect of a reduction in the scantlings of these items but, if cast steel tillers of reduced scantlings are fitted, greater attention will have to be paid to their finish. It appears to be the practice among shipbuilders, or their engineers, to bore out such holes as may be necessary and leave the remainder of the tiller in the as-cast condition. Surface irregularities, which cannot be avoided on the sharp edges of castings, are potentially dangerous and they, together with the sharp edges, should always be removed, by grinding or other suitable means, before the tiller is accepted by the Surveyor concerned.

Mr. Lockhart asks for comments on rod and chain gear mechanism with power units from colleagues with recent experience of these. There are still a number of small craft operating in the Bristol Channel area with rod and chain gear and it is fair to say that breakdowns are now a rarity ; whether this is due to the limited size and restricted services of the ships concerned, to the greater awareness of others to their responsibilities or to the annual survey requirements it would be hard to say. The annual survey requirements were originally designed for a different type of ship, operating under vastly different conditions, and they are undoubtedly onerous for the smaller type of ship to which reference has been made ; nevertheless, this does not appear to be a case for a revision of the Rules but for exercise of reasonable discretion in their application.

Mr. J. H. Nairn

Those of us in the Outports appreciate very much Mr. Lockhart's clarification of the various types and combinations of steering gears which will help us in what is often a troublesome decision as to whether the Society's requirement of two independent means of steering is or is not provided. It should save many letters being sent to Head Office in this connection which we know from experience make it difficult also for London to decide since they are not usually fully informed of the exact conditions prevailing in the particular case.

Fig. 10 on the paper illustrating hand hydraulic gear for twin rudders is not the usual type supplied in this country. It is much more usual for the power element to operate directly on one tiller only with the second tiller connected to the first by a simple control lever.

The writer is pleased that it is suggested that the Rule paragraph regarding steering gears should be revised as this has also been the writer's contention over a period of years. It may be that our experience has been extreme cases but we have vessels built in this country with the rudder directly behind the single screw which has been extended very far aft to obtain clear water and although this vessel steers like a rugby three-quarter heading for the touch-line it is practically impossibe to obtain hard over to hard over in 30 seconds in spite of a considerable restricted angle of helm. We also had twin screw coastal tankers built here for a well-known oil company with a centre line rudder which could be moved from hard over to hard over in about 10 seconds but the steering effect on the vessel was such that the owner greatly increased the rudder area in spite of the twin screws as the vessels navigated in narrow rivers and estuaries.

In view of this experience, we suggest that revision of the Rules should really deal with steering effect on the vessel rather than a time for hard over to hard over which is not the essential issue.

We would suggest a revision somewhat as follows although the figures and times which we mention are arbitrary ones which should be revised by more experienced members of the staff than the undersigned, and the idea is that Section 27 should commence with a title of "Steering Arrangements" rather than "Steering Gears" which would be followed by a title of "Rudder Operating Gear" which would include the present paragraphs from 2704 and following.

The steering arrangements we would suggest being as follows: —

STEERING ARRANGEMENTS

(a) All vessels are to be provided with two independent means of steering. One of the arrangements is to be capable of altering vessel's course through 90° in either direction within a period of 30 seconds whilst proceeding in ahead direction between three-quarters and full ahead speeds. The secondary steering arrangement is to be able to alter vessel's course through 90° in the same manner within a period of one minute whilst going ahead at ten knots or at two-thirds the full ahead speed whichever is the less.

(b) In vessels with length L over 200 ft. the above primary steering arrangement at least is to be operated by power. It is also to be demonstrated that both arrangements have positive effect on the vessel moving in the astern direction.

(c) Where hand and power hydraulic systems form both means of steering and are operated through common pipes, the system is to be capable of being controlled locally. Provision is to be made so that failure of the system between the steering wheel and the pump unit will not result in inability to steer the ship. The pump unit is to be arranged as near the tiller as possible.

(d) In twin screw vessels with suitable controls, and the consent of the owners, consideration can be given to the use of the main engines as the primary steering arrangement. The secondary arrangement being by rudder or other system completely independent of the main engines to be no less effective than secondary arrangement above when operating against the effect of one engine only in operation.

RUDDER OPERATING GEAR

Existing paragraph 2704 and continuing as existing.

MR. J. FRIZE

The progress made in steering gear design and the case for a revision of the Society's requirements are clearly presented by Mr. Lockhart. Extracts from the 1960 SOLAS Convention are quoted in the paper which should form the basis of any new Rules. This would include the trial of steering gear at maximum astern speed: unless rudder angle and time limit are specified the trial loses value. During sea trials the rudder is often not totally immersed and recorded times may be misleading on this account.

If rudder torque is to be evaluated, the diameter of rudder head required for twisting is obtainable and all steering gear items could be based on this diameter. The present tabular scantlings of rudder heads would be difficult to maintain with a simple formula, but that proposed by the Author is a lead. Has the block coefficient such a marked influence? The fast ship, generally fine, is reputedly quick to answer the small helm produced by low torque.

Rotary vane units are increasingly popular and merit the Author's full descriptions. What effect has small axial movement of the rudder head on this type? Must passenger ships be fitted with the "double-barrelled" version shown in Fig. 5? A sketch to illustrate the radial piston type mentioned on page 5 would be most helpful.

Although underpowering is rare it is nevertheless surprising to learn that the reserve can be such that no increase to the steering gear for ice strengthening is made in practice. Granted, the gear may be protected by a relief valve from shocks transmitted through the rudder head, but sufficient torque to steer the ship in ice must be available.

Reported defects suggest that materials and strength of components are satisfactory while faults in power supply show the necessity for duplicate leads. Any revision of the Rules would be more acceptable to steering gear makers, and others, if Classification Societies could be persuaded to present unified requirements.

MR. W. H. MARSDEN

During my short experience of service abroad the usefulness and helpfulness of the Staff Association Papers have been emphasised. This present paper will be read, I am sure, with great interest as the Author has given a clear and straightforward account of this rather neglected subject.

The statement made on page 4 by the Author regarding the necessity to differentiate between the rudder, the rudder stock and steering gear is worthy of emphasis; together with the remark that the forces transmitted by the stock should be protected from the steering gear. This raises the fact that most of rotary vane steering gear requires a rudder carrier due to its resilient mountings, a point which gives no problem when the builders have previous experience or where the steering gear manufacturer lives on the doorstep. It would be of interest if the Author can give information of any other type of steering gear which requires similar protection. Also can the Author advise whether it is necessary to have a type of rudder carrier which is recessed into the rudder stock, to make it effective for vertical shock loadings when resilient mountings are used for steering gear like in the vane type.

I congratulate the Author on the treatment of the method of calculating rudder torque giving essential data with clear explanation without providing anyone with the excuse to pass it to the experts or the need for a pocket computer and a shelf of reference books. This type of information shows its worth for guidance in the future as the Author states but with the owners of small shipping lines having ships built not necessarily in their own country, this type of question could be put to you to-morrow by the local representative seeking the Society's guidance.

The sections on testing and steering gear defects from reports, will be a well thumbed part of the paper and perhaps for reference it would be appropriate to mention Amendment No. 6, Instructions to Surveyors Part 3b 1957 Periodical Survey of A.E.G. Type Rotary Vane Steering Gears.

Finally, concerning tillers and quadrants and referring in particular to that part of D 2706 which states that "Tillers and quadrants are to be shrunk on or bolted to the rudder head, in addition to being secured by a key of suitable dimensions". This key of suitable dimensions varies as the country of manufacture varies and especially where steering gears are built under licence in a country where the engineering standards for such items are not the same as the country of

original design. Appreciating the immense difficulty of trying to standardise such an item can the Author give any minimum guidance notes?

Mr. N. Dienes

I should like to raise a question regarding the Astern Speed. The Author states "that in many cases the gear is in fact suitable for the maximum astern speed at reasonable angles of helm".

If we consider the case of the most commonly used steering gear, a hydraulic one, and the vessel moving ahead, the positioning of the relief valves in the hydraulic circuit are such that when the hydrodynamic torque exceeds a set valve these valves lift, allowing the rudder to move towards the midships position, reducing the torque until the valves close. However, if we now consider the case of the vessel moving astern and the limiting torque conditions are reached the relief valves again lift, but, due to the positioning of the valves this time the rudder will be forced to the hard over position and all the time the hydrodynamic torque is increasing until it reaches a maximum value when the rudder is pressed hard against its stops. The rudder can only be moved back when the vessel has been slowed down.

The following case will illustrate the previous remarks. A 500 gross tons trawler powered by a direct coupled 4SA 1800 b.h.p. diesel engine had a service speed of 15 knots. The steering gear fitted was an electric powered rotary vane unit. During the full speed astern trials it was found that if the rudder was moved beyond about 5° from the midship position the relief valves in the steering machinery hydraulic system lifted and the rudder swung round against its stops. It was estimated during these trials that the vessel's astern speed was 12·5 knots. A representative of the steering gear makers when queried about the functioning of his gear stated, quite rightly, that the steering machinery was functioning satisfactorily and developing its rated turning moment. The size of the steering gear had been specified by the shipbuilders.

Does the Author consider that in such a case one of the following steps should be taken?

(a) To endorse the classification certificate "Maximum continuous astern speed not to exceed V knots". The speed V being that at which the rudder can be controlled when running astern, or

(b) To place warning notice in a suitable position on the bridge, or

(c) To rely upon the seamanship of the Officer manœuvring the vessel.

Mr. A. K. Buckle

This is not so much a contribution in the usual sense as a series of questions which I would very much like the Author to answer in order to fill in some of the gaps in my knowledge.

1. None of the data given seems to be based on experience in vessels of much under 500 tons. Could the Author confirm whether or not this is the cast? If the answer is "no" then: —
2. As more than half the world's water borne passenger traffic and a majority of the world's seamen and watermen are carried on craft of less than 500 tons, would the Author please state if such craft require any particular treatment in view of their tendency to work continually in restricted, crowded waterways?
3. As ships fitted with Voith Schneider propellers do not require separate rudders could the Author comment on the control mechanism of these propellers in so far as the steering of the ship is concerned? Is such gear in the province of Ship or Engineer Surveyors? How does the Society test such equipment?
4. In hydrofoil craft it is sometimes necessary to restrict the rudder movement both in its limits and speed as excessive rudder angle or a too rapid an application of maximum helm can throw the craft out of control. Would the Author please comment, as fixed stops would not appear to be the answer if manœuvrability is to be maintained at slow speeds?

   In view of the very fast response of such craft to their helm and their tendency to spin about their main foils, and to heel inwards as they turn, how would the Author recommend that helm forces be calculated?
5. Could the Author please expand his list of defects a little? It would seem likely that many of the faults listed were not, in themselves, the primary cause of failure. What, for instance, is usually the cause of motors burning out and trunnion bushes falling out?
6. What are the Author's ideas regarding the use of: —
   (a) geared hydraulic motors (as an alternative to rams),
   (b) geared rod controls from the wheel to the steering engine,
   (c) the use of flexible steel wire instead of rods and chains as a connection between the steering engine and tiller,
   (d) power assisted hand steering instead of fully powered steering.

   All these systems occur on yachts and harbour craft and can be just as reliable as the systems mentioned in the paper.
7. What are the Author's recommendations for the length of the main bearings on spade rudders? I ask this as the recommended bearing length given in Association's 1957–58 Paper No. 1 seems unreconcilable with that given in the Norske Veritas Rules.

Gothenburg Ship Plans Department

Mr. Lockhart says that the most important single factor in any formula for estimating the rudder pressure is undoubtedly the speed. We require to know the speed at the rudder face and this is, generally speaking, a function of the ship speed, propeller race and wake speed. Accurate figures for the latter two speeds are not normally available and hence it is the normal practice to use empirical values for the speed when calculating rudder pressures.

The Rule requirements for rudder stock diameter are based on sea speed only and it has presumably been assumed that the increased speed at the rudder is relatively constant. As Mr. Lockhart says, there are cases when the speed of the water at the rudder is less than expected and we had such a case in Gothenburg when, due to an unusually high wake fraction, the actual wake speed was nearly one knot greater than that of similar ships of the same type. This resulted in less torque than expected on the stock and enabled us to accept an increase in rudder area without increasing the stock diameter. The ship in the example given was 180 ft. in length.

It is considered that a formula for calculating torque which differentiates between the fullness and fineness of vessels is a prudent choice and the formula for total torque given on

page 7 is to be commended for this. We see that, for instance, we would expect to have less torque on the rudder head of a full vessel than on that for a fine one and it may be mentioned here that torque readings taken lately on a 70,000 ton tanker were rather smaller than expected.

Also in connection with the torque, we notice that a centre of pressure for the ahead condition is given on page 7. In Last's paper a similar form of centre of pressure expression for the astern condition is given. For a rudder head which is subject to bending and twisting, when using those centres instead of the geometric centres as used in T 47 of the Rules to calculate the diameter, obviously differences are obtained in the results and a builder here has particularly pointed out such a case.

Regarding bolted tillers and quadrants, we agree with the doubts expressed as to the necessity of the required bolt spacing as given in T 50 of the Rules and in fact we have not been able to convince one particular tiller maker on this point. This tiller manufacturer is principally concerned in manufacturing tillers for smaller vessels and we know that they have used much smaller bolt spacing quite successfully and to obtain an equivalent Rule arrangement for ships to our class we have required bolts of higher tensile steel to be fitted.

AUTHOR'S REPLY

Before dealing separately with the contributions to the discussion it is desired to amplify the additional frictional torque suggested on page 8 of the paper in the case of rudders of the spade and underhung type.

In the case of spade rudders these increases would be 7·2 per cent for 9 in. diameter stocks, 9·6 per cent for 16 in. diameter stocks and 12 per cent for 25 in. diameter stocks. For lower pintles in underhung rudders the increases would be 3·6 per cent for a 9 in. diameter pintle and 6 per cent for a 25 in. diameter pintle.

Such increases would be dependent on the bearing surfaces and the lubrication provided. It can be shown in the case of spade rudders with gunmetal and phosphor bronze bearing surfaces and continuous lubrication, and in the case of lower pintles in underhung rudders where the length of the bearing results in a comparatively low pressure these increases could be reduced.

Some criticism has been made by the introduction of block coefficient as a factor in the formula for rudder torque. A fine lined ship appears to go into the turn more easily but it should not be forgotten that this can be related to the normal force coefficient. This coefficient is generally greater in a fine ship largely on account of the increased wake and it was thought prudent to include such an adjustment in the formula.

It may be necessary in the near future to define requirements for astern conditions. In many cases the maximum astern torque occurs around 20° of rudder angle, it should not be forgotten that the pressure on the rudder, neglecting the wake effect, can be much greater than for ahead conditions.

Reference was made in the paper to the calculation in Mr. Last's paper on Rudders and Sternframes, Appendix 2. The distance between the centroid of the rudder area from the axis of the rudder, "D" ft., used for the ahead condition could be taken as the distance from the centre of pressure of the rudder area to the axis of the rudder (ft.) for astern conditions. In the case given in Mr. Last's paper the "a" factor of ·1675 would then become ·1033 and the increase for bending would thus be 1·032 giving a diameter of $13{\cdot}25 \times 1{\cdot}032 = 13{\cdot}7$ in.

If an angle of 20° is considered the pressure in tons could be expressed as $\frac{\cdot 035}{2240} V^2 \theta \times A$

where V is the maximum astern speed in ft. per second
$= 20$

$A =$ Rudder area in sq. ft.

The centre of pressure at 20° can be taken as 0·25 breadth of the rudder from the leading edge (after edge).

To Mr. Clemmetsen

Many steering gear makers would share Mr. Clemmetsen's view that rotary vane gears are not more complicated or less substantial than the more conventional gear (presumably electro hydraulic type).

Tremendous advancement has been made in the rotary vane gear and one is now in design or even possibly in production for a gear developing 1200 ft. tons torque.

In amplification of the formula since $A \times D$ is a reasonable measure of rudder shape the values of C_t in Fig. 9 include a correction for the centre of pressure in relation to the centre of gravity plus an allowance for frictional torque.

Accepting a centre of pressure of three-eighths of the breadth from the leading edge as used by many of the makers and taking the ratio of rudder area aft to forward of 4:1 the centre of gravity would be ·30 and the centre of pressure ·18 both aft of the stock centre thus giving a correction factor of $\cdot 18 \div \cdot 30 = \cdot 60 + \cdot 05$ for friction giving ·65 as in Fig. 9.

With a ratio of rudder area aft to forward of 15:1 the centre of gravity would be ·44 and the centre of pressure ·32 giving a correction factor of $\cdot 32 \div \cdot 44 = \cdot 73 + \cdot 05$ for friction giving ·78 again as in Fig. 9. It is difficult to resolve these values with an increase in torque for friction of at least two and a half times the hydrodynamic torque as found by Mr. Clemmetsen.

The method of increasing the speed squared by 1·44 has been used by the Society as a means of assessing the pressures in lower pintles, etc., however, when dealing with the torque necessary to turn the rudder something more realistic and related to the length of the ship is required. The value of $[(V \times (1 + \cdot 01\sqrt{L})]^2$ appears to give a reasonable average through the speed and length range when compared with actual torque values found in various ships.

The astern torque is always of some consequence and perhaps not more so than in the case of ships fitted with Kort Nozzle rudders.

It would be difficult to find fault with seven knots as an absolute minimum for secondary means of steering.

The strength of tillers suggested represents present practice and while the maximum torque can occur at 35° it is still possible to calculate the strength of the tiller in the central position as suggested in the paper, and the values given are a fair interpretation of present practice.

To Mr. Stevenson

It was difficult to find any colleagues who had experience of Napier screw gear but apparently Mr. Stevenson is the exception. The personal experience described is most interesting and would be of immense value to anyone who comes across this type of gear.

We are also indebted to Mr. Stevenson for relating his further experience with hydraulic ram gear in relation to the stroke limiting arrangements. The adjustment of limit controls is not an exercise for the amateur and this is amply illustrated by the fracturing of the ram in the case mentioned.

It is a pity more of our experienced senior colleagues who must have met similar troubles could not have offered the benefit of their wisdom to younger surveyors.

To Mr. Wormald

There appears to be no record of an over energetic helmsman blowing the motor fuses, there is, however, no doubt that with the earlier hydraulic telemotor control using the larger steering wheel that a considerable effort could be transmitted to the hunting gear.

I am indebted to my electrical engineering colleagues for the explanation that the omission of overload protection arranged to trip the electrical supply follows the established practice of avoiding the loss of this fundamental service. While it intended that temporary overloads should not necessarily put the gear out of action the alarm is provided to warn the ships personnel of the existence of such a condition.

In the case of the tanker quoted by Mr. Wormald some support is advocated for the use of an alternative source of power for the emergency gear. With the present efficiency of the electro hydraulic gear it would, however, be difficult to make such a separate means of steering a rule requirement.

The suggested amendments in the paper to paragraphs 2701–2705 by the introduction of a minimum angle of rudder movement for the secondary means of steering is certainly not intended to be retrograde. It is sometimes forgotten with electro hydraulic gear having two motors and two pumps that the rules are satisfied if the rudder can be moved from hard over to hard over in 30 seconds with both pumps and motors running and that no time limit is stated for the secondary means of steering, thus if one motor and one pump took 60 seconds to move the rudder to hard over no exception could be taken to such a performance.

It should not be forgotten that the emergency steering is not necessarily expected to perform the duties of the primary gear. It is therefore thought reasonable that some acceptable standard can be brought speedily into action the requirements stated under paragraphs 2701–2705(b) would certainly satisfy the requirements of the 1960 Safety Convention. The requirements stated for tillers and quadrants are largely a translation of present practice. Mr. Wormald is to be thanked for pointing out the danger of abusing cast tillers, the avoidance of surface irregularities and rough edges is certainly advisable.

It is interesting to note that rod and chain gear with power units are still performing admirably in the Bristol Channel area. Even though the service of such ships is limited perhaps the gears are performing their steering functions more continuously than many ocean-going types, and no small account of the continuing success is due to adequate surveying and upkeep.

To Mr. Nairn

There are, as pointed out, various alternative arrangements of Fig. 10 and control of the second tiller by a cross lever is not uncommon. If a preference was expected the gear illustrated in Fig. 10 has the superiority of less frictional resistance. Obviously, the position of the screw relative to the rudder has a marked influence on the slip stream and thus on the pressure exerted on the rudder. The example of this given by Mr. Nairn may be one where the effect of lack of deadwood caused by the absence of a rudder post or horn has a marked effect on the rudder pressure. In such cases the shape of the rudder is of fundamental importance and by the description of steering conditions given it would appear that the proportion of area forward to aft of stock centre could have been amended. It also appears that the torque developed by the gear was deficient but this could have been greatly accentuated by a high frictional torque.

In the case of the twin screw coastal tankers quoted it is apparent that the rudder area was deficient in the first instance. This may also have been agitated by incorrect proportioning of the balance.

It is difficult to be pedantic about steering arrangements, in most cases if the gear performs effective steering in moving to hard over in the required time the turning circle will be satisfactory.

Regarding Mr. Nairn's suggested revision of the Rules as per paragraph (A), while 90° of travel may be desired by some owners for their particular case of manœuvrability it is not thought that this should be made a requirement. It can be demonstrated in most rudders that such a travel is only effective at speeds less than the maximum and the torque required will be less than that for a 65° travel at full speed.

The suggestion in paragraph (D) that the main engines be used as the primary means of steering in twin screw ships is not one which would have general appeal. There is no doubt that a ship can be manœuvred to some extent by this means but not sufficiently so to eliminate the use of the rudder or rudders for both main and auxiliary steering.

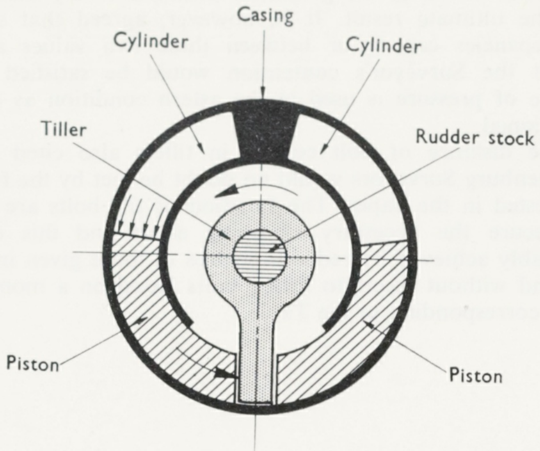
To Mr. Frize

It is agreed that if any standard is to be laid down for astern trials that some helm angle would require to be stated but when going astern at full speed the rudder should not necessarily turn through 65° in any specific time. It might, as already stated, be reasonable to expect the gear to move the rudder from 20° on one side to 20° on the other and *vice versa* when going astern at full speed.

Mr. Frize suggests that rudder heads could be related more closely to a twisting moment probably based on the centre of the pressure. There is justification for this thought particularly in the case of balanced rudders but as pointed out in the paper the stock has to perform duties more onerous than those required by the steering gear, and no radical departure from present standards could be envisaged.

It is not a requirement of the Society to fit dual chambers in rotary gears in passenger ships but common agreement on this particular item will probably be resolved when the 1960 Safety Convention requirements are implemented.

A sketch of the radial piston type gear is given below.



Radial Piston Steering Gear Unit.

Regarding navigation in ice large angles of helm should seldom be necessary when operating under such conditions and as already stated the steering gear apart from specific items already mentioned in the paper is protected by the relief valve.

The plea for more unified requirements for steering gear should be largely met as and when the requirements of the Convention become operative.

To Mr. Marsden

Rudder carriers are now common to most gears and it is interesting to note that most steering gear makers now favour the flat-topped type with an adequate vertical bearing to resist side thrust. The steering gear maker can generally select the type and positioning of the bearing most suited to the gear. In some of the smaller types of rotary gear the bearing is incorporated in the unit and in these cases the maker states that a bearing is unnecessary, this, however, requires further consideration in the case of spade or similar type rudders where the stock is subjected to bending.

Recessing of the stock for the bearings should generally be avoided as horizontal and vertical surfaces can be provided without resort to necking the stock.

Mr. Marsden's comments regarding the application of the rudder torque are appreciated and it is hoped that the suggested value might be of some use in giving advice to owners, etc., who might request such guidance from the Society's Surveyors.

The keying of quadrants can, as stated, vary considerably. If the key is to be related to the stock at the same stress, the sectional area would be expressed as $\cdot 39d^2$ sq. in. where d is the diameter of the stock in inches. This value, however, is perhaps more correct for fast-moving shafts and if modified to about $\cdot 33d^2$ gives a fair average for tiller security.

To Mr. Dienes

The case presented is interesting and to some extent exceptional and is related almost exclusively to astern conditions.

Although no particulars of the case are available it would be helpful if Mr. Dienes could forward these. The torque values derived from Fig. 9 in the paper could be reduced in the case of some balanced rudders and no doubt a gear based on such a reduction would move the rudder from hard over to hard over at full ahead speed within the specified time limit. For example, in the case related by Mr. Dienes, if the rudder were balanced with two-thirds of the area aft and one-third forward of the stock the torque required for ahead conditions (15 knots) could be exceeded at about 10° of helm when going astern at a speed of 12·5 knots.

The torque values derived from Table 9 at an ahead speed of 15 knots would be generally satisfactory for about 10 knots astern with the rudder moved to hard over. There appears to be makers of the small type of gear who concern themselves very little with astern conditions and probably the case quoted is one of these.

Suggestions (a) and (b) by Mr. Dienes could hardly be supported as no standard is at present laid down in the rules for astern steering conditions.

It would probably be wise to ascertain the speed at which the existing gear can manœuvre the ship astern which would appear to be about 7 or 8 knots in the present case, and make the ship's personnel aware of this fact. No doubt the owner would also soon be informed. It would be further prudent to acquaint the owner that where any particular astern requirements are necessary these should be specified and the rudder and steering gear could be designed in the first instance on this basis.

To Mr. Buckle

The data given are generally relevant to ships under 500 tons. The steering arrangements including rudder area, etc., of ships which operate in busy restricted water always demand this aspect as a criterion of consideration.

In the case of Voith Schneider propulsion the control mechanism should be examined and be tested under working conditions. As with all gears the combined experience of ship and engineer surveyors are applied as necessary.

It was not intended in the paper to deal with specialities, e.g. hydrofoil craft. A hard over in such craft might be 10°, the fact that no stops are provided could be attributed to the fact that the steering of such craft at varying speeds is the duty of a specialist. Before attempting to calculate pressures associated with such craft a great deal more data will require to be collected.

The defects listed were intended to illustrate where troubles can occur and it can be said that many of these were not due to any overloading. A motor can burn out while driving any form of mechanism and in many cases the cause is neglect. Trunnion bushes in some of the older type of ram gear were not properly secured and experience has largely remedied this defect. It may be, however, that having brought this to the notice of colleagues particularly in surveying older ships the security of such bushes would receive their attention.

The types of gear mentioned by Mr. Buckle under (6) can all be accepted for particular cases. It is worth mentioning in the case of steel wire ropes that these have been generally confined to very small craft as the stretch of the wire can make steering difficult and it is not an effective equivalent to rods and chains.

The length of main bearings in spade rudders must be considered in relation to the bearing pressure and to further adjacent supports. In most cases it has been found where an effective upper bearing is fitted a length of about one and a quarter to one and a half times the stock diameter is adequate for the lower bearing. At least one case is known where a bearing length of four times the stock diameter produced a

frictional torque which was excessive for the gear and the bearing arrangement was modified to reduce the frictional resistance.

To the Gothenburg Ship Plans Department

The department is to be thanked for their valuable and reasoned contribution. The example given clearly shows the danger of assuming a constant empirical increase for wake effect. There is little doubt that a great deal of data could still be collected. It is gratifying to note that the department concurs in the suggestion to apply a correction for block coefficient and it is believed that this can be justified by experience.

In the case quoted by Mr. Last the effect of the centre of pressure in lieu of the geometric centre has very little effect on the ultimate result. It is, however, agreed that sizeable discrepancies can occur between these two values and no doubt the Surveyor's contention would be satisfied if the centre of pressure is used in the astern condition as already mentioned.

The distance of bolt centres in tillers also cited by the Gothenburg Surveyors would no doubt be met by the formula suggested in the paper. The function of the bolts are chiefly to secure the necessary clamping effect and this can be probably achieved by reduced centres to those given in Table 50 and without resort to fitting bolts based on a moment of area corresponding to the Table.

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